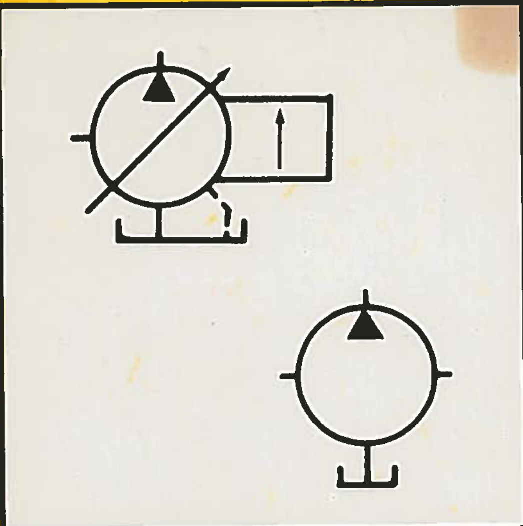
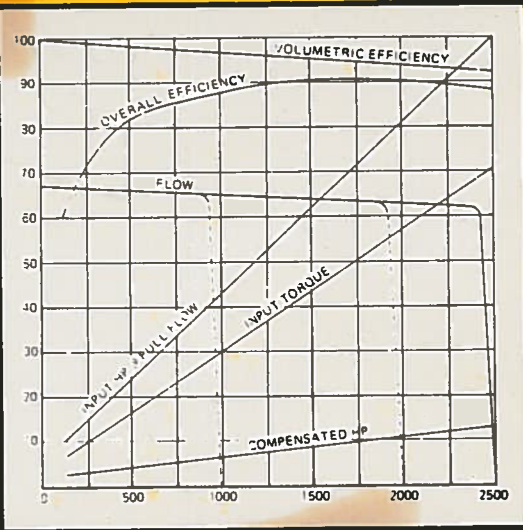
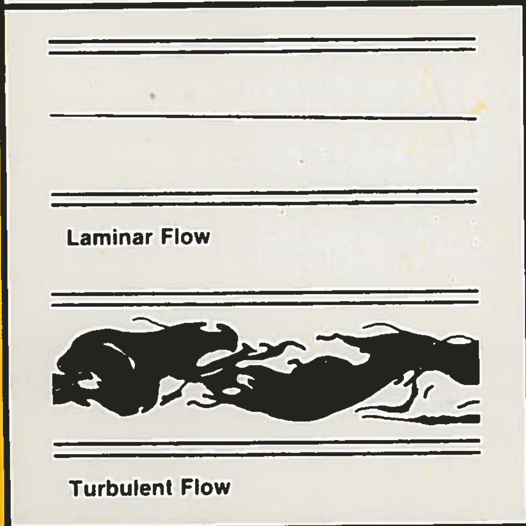
Analyzing Hydraulic Systems

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$$C_v = Q\sqrt{\frac{Sg}{\Delta P}}$$

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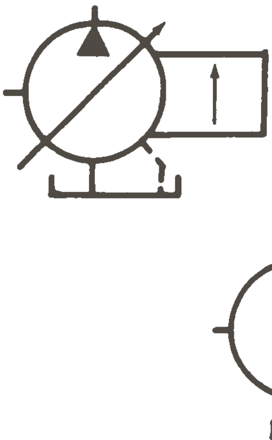
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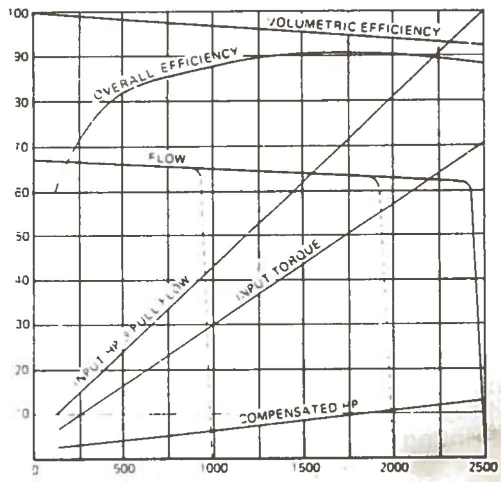
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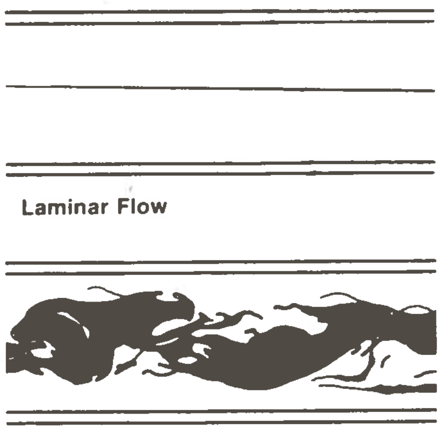
Analyzing Hydraulic Systems

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Laminar Flow

Turbulent Flow

$$C_v = Q\sqrt{\frac{Sg}{\Delta P}}$$

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ANALYZING HYDRAULIC SYSTEMS

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PREFACE

When designing a machine or system that takes power, then converts, manipulates and transmits it to perform work under controlled conditions, certain rules can apply. If the rules are strictly adhered to, a more efficient design will result.

An analysis of the fluid power components that perform the power transmission and control task and their relationship to each other is a prerequisite for increasing system efficiency. To carry out an efficient analysis, the components and their interrelationships must be considered as we set up a circuit or schematic diagram. Each component is represented by a standard symbol and it is this symbol that represents the function of the component. The inter-connecting lines between components establish their relationship as to sequence of operation as well as their circuit relationship.

Once created, a schematic diagram can be readily transformed into a physical system. However, it's physical system can be only as effective as the analysis, thought processes and ingenuity applied to the conceptual design. It takes a good conceptual design to achieve a unit which will continue to function efficiently. Errors in circuit analysis and design in the selection of components will certainly show up in the operation of the machine. Inefficient operation, excessive heating, shocks, noise, vibration and a need for frequent maintenance are a few of the symptoms that can result from poor design procedure. These problems show up rather quickly after initial startup. Remember, circuit design errors, no matter how small, tend to be compounded, and can result in expensive and excessive downtimes and even more expensive re-design and/or repair.

It is the purpose of this text to help the student to become proficient in the analysis and evaluation existing in hydraulic systems. With this knowledge, the student is prepared for advance study in basic hydraulic system development. A working knowledge of the elementary laws of physics is assumed at the onset of the book. Components and circuitry are analyzed from an energy conservation standpoint. Efficiency and heat generation in the system is stressed.

We hope the student will find the course of study logical and easily understood. Future editions will include additional information and revisions suggested by those who use this text. Your comments and suggestions are cordially requested.

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CHAPTER 1

THE HYDRAULIC SYSTEM

Many textbooks state that the ultimate objective in fluid power analysis is the transmission of power. From that point on the definition of power seems to fade into the background. Instead, discussions dwell on the operation of components, transfer functions, development of flow, and the application of functional blocks. Details concerning the actual problems concerning transmission of power are circumvented.

Certain hydraulic texts discuss what a component does and how it functions, rather than how it's power transmission characteristics fit into the total power system, as a potential power wasting circuit element.
Typical examples are:

a. Directional valves are described in the functional contexts of open center vs. closed center designs.

b. Pumps are compared as fixed vs. variable.

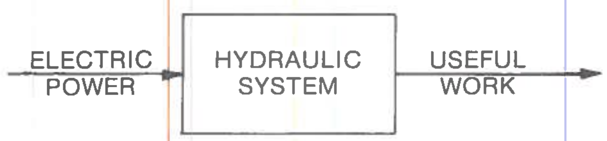
c. Pressure controls are shown in sequencing circuits and pressure reducing circuits.

d. Flow controls are either metered-in, metered-out, or bleed-off.

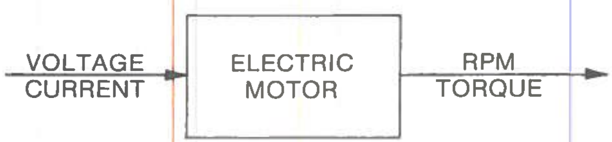
e. Cylinders are viewed in overrunning load vs. static pressing.

All of these involve functional descriptions. The use of pressure controls, for example, is often treated as an isolated type of circuit element without relating to other control functions in the fluid power system. Where a hi-lo system is incorporated, the balance of the system is selected to fit this hi-lo system portion rather than the other way around. Efficient power transmission can be achieved only when the component is selected to fit the fluid power design needs of the total machine or process to be controlled.

Every component in a hydraulic system has a particular relationship to power distribution. As a matter of fact, we may construct a block diagram showing how power is changed as it moves from one point to another in a hydraulic system.

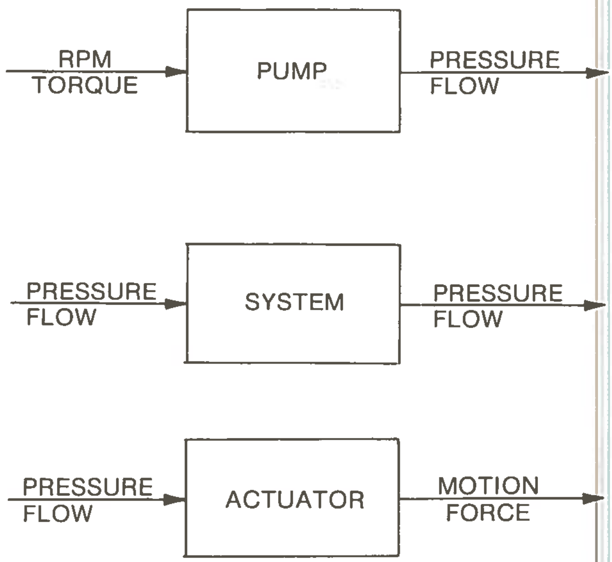


We will start with the input power to the hydraulic system. No matter whether we select an internal combustion engine, an electric motor, or even water power, this device will deliver a particular speed (RPM) and a torque. Remember, we are working with power, therefore, speed and torque must represent power and it does. Power is directly proportional to speed and torque.



This speed and torque will then be transformed into

hydraulic power. To do this, we will input speed and torque to the hydraulic pump. The pump, whether it be a fixed displacement or variable displacement, will deliver a flow rate at a particular pressure. To convince ourselves that the pump is delivering power, let us look at the following.



Power is proportional to force and velocity. Since force equals pressure times area, this may be substituted for force. If we combine velocity and area, we obtain flow. In the resulting equation, flow and pressure is proportional to power.

Now as we travel through the components situated after the pump in the system, we see that flow and pressure are both inputed and outputed. The only variance will be the magnitude of the power compared to power out.

The final component in the system is the hydraulic cylinder or motor. Here it accepts a flow and pressure delivered and reconverts it into a motion and a force. Thusly, we have completed the cycle-power in yields power out.

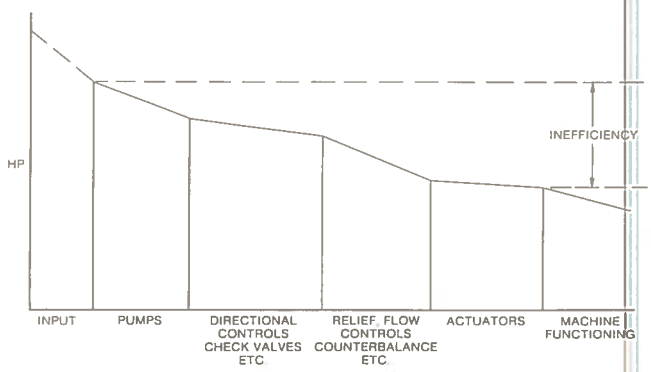
Of course, other components come into play when designing a system. These elements, controls, filters, reservoirs, etc., are necessary in order that the system function correctly and have a sufficient life.

HYDRAULIC SYSTEM EFFICIENCIES

The efficiency of a hydraulic system is dependent on several factors. A few examples are:

1. The efficiency of the hydraulic components.
2. The efficiency caused by individual components interacting with one another while the system is in operation.
3. Transmission losses.

A qualitative idea of the inefficiency of various components can be seen in (Figure 1-1). As can be seen from the figure, flow controls and some pressure controls contribute to excessive power consumption. Each one of the areas will be covered in depth. The best area to start is the cylinders and motors. This will give us a good foundation for the rest of the text.



(Figure 1-1)

CHAPTER 2

CYLINDERS & MOTORS

In almost all power systems found on this earth, the desired end result is the doing of some type of work. To do mechanical work, a conversion of energy must take place. Just as we transform the energy of gasoline into rotary motion in a gasoline engine, we must transform the power delivered by a pump, a flow rate and pressure into a form capable of delivering useful work.

The pump is delivering a certain flow rate (measured in gallons per minute — GPM or litres per minute — LPM) at a particular pressure determined by the system. This flow rate and pressure must be converted into useful work. The most commonly employed converting units for hydraulic circuits are cylinders and motors. Cylinders will be discussed first.

LINEAR POWER TRANSMISSION — CYLINDERS WITH NO MOTION

A fluid power cylinder accepts fluid pressure and converts it into rotary or linear motion with an associated force or torque. Since most cylinders are used to produce linear motion, these will be the ones considered here.

There are three important formulas which will be used in this section. The first formula has to do with hydrostatic force transmission.

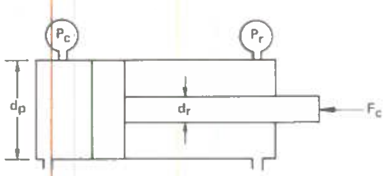
Force is equal to a pressure being applied to a particular area. The formula describing static action is:

FORCE = PRESSURE X AREA

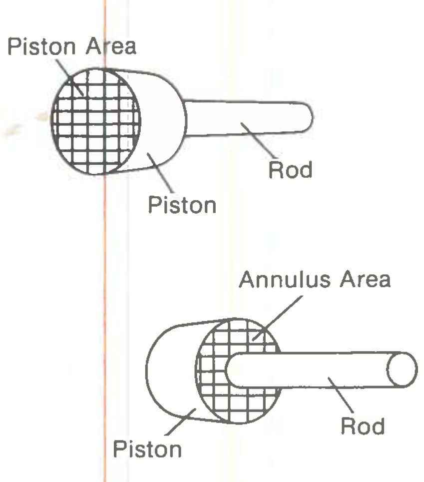
F = P x A (2-1a) where F is pounds force (#f)
P is PSI
A is in square inches (in^2)

$$F = \frac{P \times A}{0.1} \quad (2\text{-}1b)$$

Where F is Newtons (N)
P is bar
A is cm^2



(Figure 2-1)



(Figure 2-2)

In order to derive the complete force equation for a cylinder with no motion, consider (Figure 2-1). This cylinder has a pressure Pc acting on the piston area. A pressure Pr acts on the annulus area. "F", which is an external force, keeps the rod in compression. To find the force developed by Pc, we must first find the area of the piston.

The area of a circle (in this case the piston) equals π (3.14) times the diameter squared divided by 4 or:

$$Ap = \text{Piston Area} = \frac{\pi\ dp^2}{4} = \frac{3.14\ dp^2}{4} = 0.7854\ dp^2$$
(2-2)

dp = diameter of piston (in) (mm)

The pressure Pr acts on the annulus area Ae. The annulus area is the area of piston minus the area of the rod. Consider (Figure 2-2). Hydraulic pressure acts on the piston, however, it cannot act on the area to which the rod is attached. This means the area we need to find is Ae or the area of the piston minus the area of the rod.

$$Ae = Ap - Ar = \frac{\pi}{4}(dp^2 - dr^2) = 0.7854\ (dp^2 - dr^2) \text{ for U.S. units}$$

(2-3a)

OR:

$$Ae = 0.007854\ (dp^2 - dr^2) \text{ For SI units}$$

(2 - 3b)

Where Ae = piston area (in^2) (cm^2)
Ar = rod area (in^2) (cm^2)
dp = diameter of piston (inches) (mm)
dr = diameter of rod (inches) (mm)

Therefore, equation (2-1) will become (for a compressive load)

$$Fc = Pc \times Ap - Pr \times Ae$$

(2-4)

Where Fc = the load in pounds force or (neuton ÷ 10)
Pc and Pr are in PSI or bar
Ap and Ae are in square inches or square centimeter

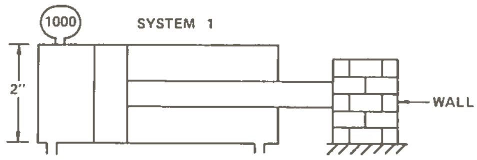
If the load is tensile (or the rod is pulling on the load), the formula will become:

$$Ft = Pr \times Ae - Pc \times Ap$$

(2-5)

EXAMPLE: (2 - 1)

System #1 has 1000 psi acting at the cap end of a cylinder of 2" diameter (Figure 2-3). The rod is stopped by a concrete wall so there is no piston motion. We want to find the force exerted by the rod on the wall.



(Figure 2-3)

SOLUTION:

Formula (2-4) may be employed because the load is compressive. Because there is no motion, zero pressure at the rod end of the cylinder would be a good assumption. The formula becomes:

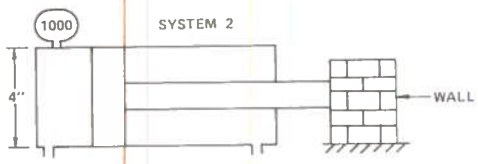
$$Fc = Pc \times Ap - 0 \times Ae$$

$$Fc = Pc \times Ap$$

Ap can be found by using formula (2-2).

By inserting the corresponding numbers, we may calculate the area equal to:

$$\text{Area of piston} = Ap = \frac{(3.14) \times (2)^2}{4} = .7854 \times (2)^2$$



(Figure 2-4)

$$Ap = \frac{3.14 \times (2) \times (2)}{4} = 3.14 \text{ square inches}$$

Since the pressure is known and read at the gage to be 1000 PSI, the force is calcualted using formula (2-4)

$Fc = Pc \times Ap$

$Fc = 1000 \text{ psi} \times 3.14 \text{ in}^2 = 3140$ pounds force acting on the wall.

EXAMPLE: (2-2)

Now consider a second case when the diameter of the piston is doubled (system 2, Figure 2-4).

The rod once again is stopped by the wall and we need to know what force is exerted by the rod on the wall.

We will employ formula (2-4) again and find Pr equal to zero. The next step is to find Ap with formula (2-2).

$$Ap = \frac{\pi dp^2}{4} = \frac{3.14 \times (4) \times (4)}{4} = .7854 \times (4)^2$$

$Ap = 12.56$ square inches

Inserting $Ap = 12.56 \text{ in}^2$ into formula (2-4) we find:

$Fc = Pc \times Ap = 1000 \text{ PSI} \times 12.56 \text{ in}^2$

$Fc = 12560$ pounds force

It should be noted that by doubling the diameter we have increased the force, four fold (the square of the ratio of diameter compared).

The next parameter which would be of interest is the rod speed. But since the piston is stationary, the rod speed is zero. Consequently, the horsepower delivered is zero because horsepower is defined as work per period of time. Since no motion exists, no work is done. Therefore, work horsepower is zero.

Important points in this section for a stationery piston:

1. A larger area for a given pressure gives a larger force.
2. Work done is zero.
3. Horsepower delivered is zero.

CYLINDERS IN MOTION

When cylinders are in motion, we are interested in finding the pressure in the cylinder, the speed of the rod, and the horsepower generated.

The pressure in the cylinder can be calculated from formula (2-4) or (2-5). Either Pc or Pr is given in these cases.

The rod speed can be obtained by using the formula that relates the flow into a cylinder to the cross-sectional area times the piston speed.

$$Q\ (GPM) = \frac{A\ (in^2)}{231} \times V\ (in/min)$$

(2-6a)

Where Q is in GPM
A is in IN^2
V is in in/min.
231 in^3/gallon

The S.I. units for the same formula would be:

$$Q\ (LPM) = A(cm) \times V(m/sec) \times 6.0$$

Where Q is in liters/min
A is in $centimeters^2$
V is in meters/sec.

The area (A) in this case, can be the piston or annulus area depending on which side the fluid enters. The horsepower delivered by a cylinder is calcuated through the use of the formula:

$$\text{Input HP} = GPM \times PSI \times .000583$$

(2-7a)

OR

$$\text{Output HP} = \frac{V\ (in/min) \times F\ (pounds\ force)}{396{,}000}$$

(2-8a)

Where V is rod speed (in/min)
F is in pounds force

S.I. equivalent would be:

$$\text{Input Power (KW)} = \frac{Flow\ (LPM)}{600} \times Pressure\ (bar)$$

(2-7b)

$$\text{Output power} = \frac{Speed\ (m/s)}{1000} \times Force\ (N)$$

(2-8b)

Formula (2-7) deals with the hydraulic power that can be accepted by the cylinder. GPM is the flow rate going into the cylinder and PSI is its associated pressure. The efficiency of linear actuators depends on many things. Some of them being:

1. concentricity of piston in bore
2. piston seal friction
3. rod seal friction
4. concentricity of rod to rod gland
5. use of cushions
6. size of internal flow passages
7. alignment of cylinder with work
8. flow losses in lines leading to and from cylinder
9. the stroke length
10. mounting attitude (horizontal or vertical)

If a cylinder is designed and manufactured correctly, one and four can be disregarded. Two or three are a function of the type of elastomer used in the seals and the configuration of the aforementioned component. It can most be attributed to the squeeze built in by the cylinder designer. Surface finish and characteristic are also important. In small bore cylinders, under 2 inches, seal friction may be high. However, for the most part, efficiency of larger bore hydraulic cylinders is high. Because of this, throughout this text we will consider the efficiency of a cylinder to be 100%. However, there are situations where the efficiency of a cylinder must be considered in the circuit design.

Formula (2-8) is the mechanical equivalent of the HP formula: V is the rod speed in inches/min and F is the force delivered.

Let us use the aforementioned formula's in the following problem.

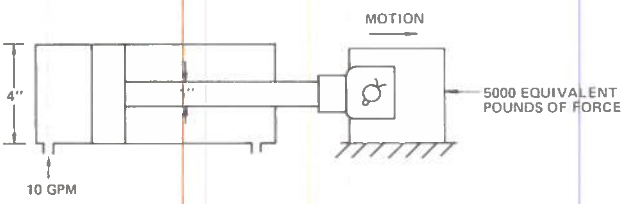
If the efficiency is considered to be 100%, formula (2-7 and 2-8) are equal.

EXAMPLE: (2-3)

A pump delivers 10 GPM (37.85 LPM) to a 4 inch (101.6 mm) diameter cylinder moves a load requiring 500 pounds thrust. (22240N)

1. What is the pressure to extend the cylinder?
2. What is the rod speed?
3. What is the HP developed?

(Figure 2-5)



(Figure 2-5)

SOLUTION

First, we will find the pressure at the gage. Employing (2-5) we find:

Fc = Pc x Ap - Pr x Ae and Pr = 0

Then . . . Pc x Ap = Fc or Pc = Fc/Ap

Area of the piston (Ap) is determined from (2-2) which is:

$$Ap = \frac{\pi\ dp^2}{(4)} = \frac{(\pi)\ (4)\ x\ (4)}{4} = 12.56\ in^2\ (81\ cm^2)$$

Since F is equal to 5000 pounds force:

$$Pc = \frac{5000}{12.56} = 398\ psi = \frac{22240N\ x\ 0.1}{81\ cm^2} = 27.45\ bar$$

Secondly, we find rod speed. We use formula (2-6). It reads:

$$Q = \frac{A \times V}{231} \qquad \text{OR} \qquad V = \frac{Q \times 231}{Ap}$$

Where Q is the flow rate — 10 GPM (37.85 LPM)
Ap is the area of the piston — 12.56 in^2 (81 cm^2)

Since fluid enters the cap end, Ap is used,

$$\text{Rod Speed} = V = \frac{10 \text{ GPM} \times 231}{12.56 \text{ in}^2} = \frac{184 \text{ inches}}{\text{min}} (0.0778 \text{ m/sec})$$

The hydraulic horsepower can be calculated from equation (2-7):

Input HP = GPM x PSI x .000583
Input HP = 2.32 HP (1.731 KW)

Where GPM = 10
PSI = 398

(Input power = 37.85 (LPM) 600 x 27.45 bar = 1.732 KW)

To check the HP equation, we may use the mechanical HP equation (2-6). Here we have:

$$\text{Output HP} = \frac{V \times F}{396{,}000}$$

Where V is the rod speed (in/min)
F is force (pounds force)

$$Hp = \frac{184 \text{ (in/min)} \times 5000 \text{ (pounds force)}}{396{,}000}$$

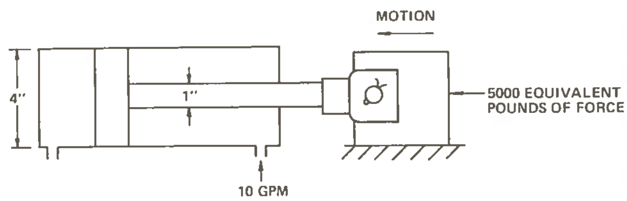
Output HP = 2.32 (1.731 KW)

$$\text{(Power out} = \frac{0.0778 \text{ (m/s)} \times 22240\text{N}}{1000} = 1.731 \text{ KW)}$$

The equations check. Both yield the same power.

EXAMPLE: (2-4)

Let us consider the 10 GPM flow to be entering the rod end of the cylinder (2-6)



(Figure 2-6)

First, we would like to know the pressure. The pressure may be calculated with formula (2-5) because it is tensile load.

Equation (2-5) reads: Ft = Pr x Ar - Pc x Ap

Since Pc = 0 the equation reduces to:

Ft = Pr x Ae

Therefore:

$$Ae = \frac{\pi(4x4) - (1x1)}{4} = \frac{\pi(16-1)}{4} = \frac{\pi(15)}{4} = 11.78 \text{ in}^2 \text{ (75.9 CM}^2\text{)}$$

$Ae = 75.9\ cm^2$

The pressure can now be calculated from (2-5) and is:

$$Pr = F/Ae = \frac{5000}{11.78} = 425 \text{ psi } (29.3 \text{ bar})$$

As you can see, it takes more pressure to return the load then extend it because of the respective decrease in area.

The rod speed may be calculated using equation (2-6). Since the fluid is entering the rod end, the annulus area is used in the formula:

$$V = \frac{Q \times 231}{Ae} = \frac{10 \text{ GPM} \times 231}{11.78} = \frac{196 \text{ inches}}{\text{min.}} \ (0.083 \text{ m/s})$$

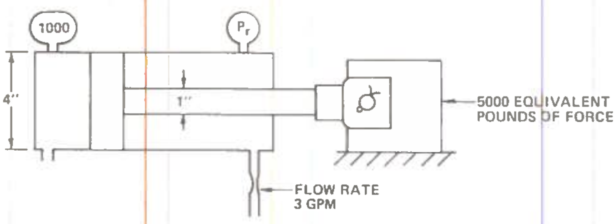
The rod speed in this case has increased and so has the rate of work performed. This is due to the same quantity of oil entering, but the available volume is smaller.

The horsepower developed by the actuator is:

Input HP = GPM x PSI x .000583
Input HP = 10 GPM x 425 PSI x .000583 = 2.47 HP (1.842 KW)

The HP has increased because the pressure needed to accomplish the work has increased.

EXAMPLE: (2-5)



(Figure 2-7)

The final case that will be considered is one in which the flow path leaving the cylinder is choked, restricting flow. (Figure 2-7)

The restriction allows 3 GPM out of the rod end where a pressure of 1000 psi exists at the cap end. The pump will deliver whatever GPM is necessary to fill the cap end of the cylinder. (1000 PSI being the relief valve setting) What is:

1. the pressure Pr?
2. the rod speed V?
3. the flow rate into the cylinder?
4. the useful horsepower?

SOLUTION:

Since there is a compressive load, use formula (2-4). Since there is a restriction in the return line of the cylinder, a "back pressure" is developed on the annulus area. This "back pressure" will be dependent on the load.
Writing formula (2-4) we find:

$Fc = Pc \times Ap - Pr \times Ae$

Where Pc = 1000 PSI & is given
Fc = 5000 pounds force

Formula (2-2) yields Ap

$$Ap = \frac{\pi\ dp^2}{4} = 12.56\ in^2$$

Formula (2-3) yields Ae

$$Ae = \frac{\pi\ (dp^2 - dr^2)}{4} = 11.78\ in^2$$

Inserting these numbers into (2-4) we find:

$Fc = Pc \times Ap - Pr \times Ae$

$5000 = 1000 \times 12.56 - Pr \times 11.78$

$12560 - Pr \times 11.78 = 5000$

$Pr \times 11.78 = 12560 - 5000$

$$Pr = \frac{7560}{11.78} = 641\ psi$$

This type of back pressure on a cylinder will always exist when a pressure compensated restriction (flow control counterbalance valve, etc.) is placed to meter flow out of the system. The maximum pressure ever seen at the metered-out side of the piston will be when no load is present. Consider no load in the above example. F will be zero with Pc = 1000, Ae = 11.78 in^2 and Ap = 12.56 in^2.

$Fc = Pc \times Ap - Pr \times Ae$

$1000 \times 12.56 - Pr \times 11.78 = 0$

$$Pr = \frac{12560}{11.78} = 1066\ psi$$

This means with 1000 psi at the cap end, the maximum pressure existing at the rod end is 1066 psi. This process is called intensification. This becomes exceedingly important when rod diameters get large. Pressures of a high intensity over the rated cylinder pressures can cause rod seals to fail prematurely.

The next parameter we must find is the rod speed. Equation (2-6) may be used.

$$Q = \frac{A \times V}{231}$$

Where A = Ae annulus area
Q = is the flow rate out of the restriction
V is the rod speed

$$V = \frac{231 \times Q}{Ae} = \frac{231 \times 3}{11.78} = 58.83\ \frac{inches}{min.}$$

The flow into the cylinder can be obtained from equation (2-6) also. However, in this case, V is known and A becomes Ap, the piston area.

$$Q = \frac{Ap \times V}{231} = \frac{12.56 \times 58.83}{231} = 3.2 \text{ GPM}$$

The HP can be found using equation (2-7). We would use:

Input HP = GPM x PSI x .000583

However, since there is pressure on both sides of the piston, it may be difficult to use this equation. Equation (2-8) may be easier to handle.

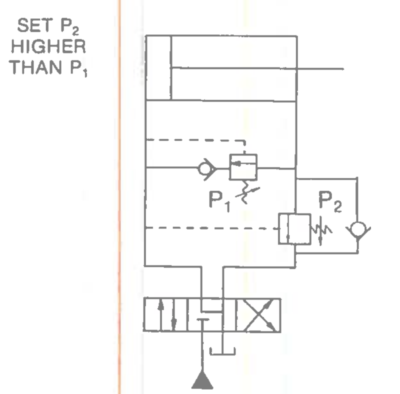
$$\text{Output HP} = \frac{V \times F}{396{,}000}$$

Where V = 58.83 in/min
F = 5000 lbs. force

$$\text{Output HP} = \frac{58.83 \times 5000}{396{,}000} = .742 \text{ HP}$$

It is easier to employ equation (2-8), in applications dealing with pressures acting on both sides of a piston. Remember equation (2-7) equals (2-8) because the efficiency is at an assumed 100%.

SPECIAL APPLICATIONS — REGENERATION



(Figure 2-8)

Regeneration or use of a differential cylinder gives one the ability to rapidly advance a cylinder to a work load, without increasing pump flow.

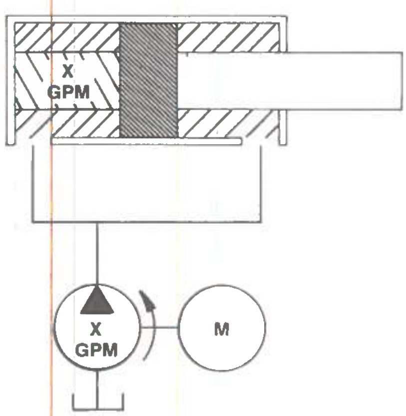
The cylinder is extending by regeneration as shown in (Figure 2-8). Both sides of the exposed piston are equal pressures. It would appear that applying the same pressure to both sides of a cylinder would cause it not to move. However, this is incorrect. Because there is a difference in areas between the cap and annulus, a net force is developed. The magnitude of this force may be derived from equation (2-4).

Equation (2-4) reads:

Pc x Ap - Pr x Ae = Fc

Where Fc is the developed force.

In a regenerative system, Pc and Pr are equal because they are connected to the same line. (Figure 2-9).



(Figure 2-9)

Therefore:

Pc = Pr = P

P x Ap - P x Ae = Fc

OR

P x (Ap - Ae) = Fc

From equation (2-3) we find:

Ae = Ap - Ar

Substituting (Ap - Ar) for Ae in (2-9)

$P \times (Ap - Ar) = Fc$

(2-10a)

$P \times Ar = Fc$ For U.S. Units

$\frac{P \times Ar}{0.1} = Fc$ For S.I. Units

(2-10b)

Where P = Pressure (psi) (bar)
Ar = Rod Area (in^2) (cm^2)
Fc = Compressive force (#F) (N)
dr = Diameter of Rod (in) (MM)

$Ar = \frac{\pi dr^2}{4} = .7854dr^2$ (in)

$Ar = (cm^2) = 0.007854\ dr^2$ (mm)

(2-11)

Equation (2-10) should always be applied when a regenerative circuit is used.

Next, we need to know the speed at which the cylinder extends. As the piston moves, the fluid which is discharged from the rod end of the cylinder is transferred to the cap end adding to the pump flow. The pump must only fill the volume which is being uncovered as the rod moves. (Figure 2-9)

Since the rod volume is the only volume that the pump must fill, the equations for rod speed will be changed to reflect this fact. Formula (2-6) reads:

$Q = \frac{A \times V}{231}$

Where Q = flow (GPM
A = area (in^2)
V = rod speed (in/min)

When a regenerative circuit is extending, the formula will become:

$Q\ extend = \frac{Ar\ (in^2) \times V\ (in/min)}{231}$

(2-12)

$Q\ (LPM) = Ar\ (cm^2) \times V\ (m/s) \times 6.0$

When the cylinder is retracted, the formula developed previously for a retracting cylinder will be used.

It should be kept in mind that when using a regenerative circuit, rod speed increases, but cylinder force decreases. This is due to the fact that system pressure exists on both sides of the piston.

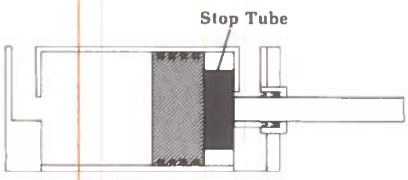
STOP TUBE

A stop tube is a metal collar which fits over the piston

rod. A stop tube keeps the piston and rod gland bushing separated when a long stroke cylinder is fully extended.

Since it is a bearing, a rod gland bushing is designed to take some loading when supporting the rod as it extends or retracts.

Along with being a bearing, a rod gland bushing is also a fulcrum for the piston rod. A stop tube in effect protects the rod gland bushing loading at full extension between both piston and bushing. (Figure 2-10)



(Figure 2-10)

Steel rods of long stroke cylinders sag just because of their weight. A $\frac{5}{8}$" (15.875 mm) diameter piston rod weighs 1 lb. per foot (1.488 kg./m) and will sag over 1 in. (25.4 mm) at the center of a 10 ft. (3.048 m) span.

A stop tube is used to separate bushing and piston when the rod is extended. This reduces the load on the rod gland bushing.

SELECTING A STOP TUBE

The selection of a stop tube depends on the basic length and the maximum thrust developed. Basic length is a function of the actual stroke of the cylinder and the type of cylinder mounting. The formula for basic length is:

Basic stroke length (in) = stroke (in) x stroke factor

Stroke factor can be determined from (Figure 2-11). As seen from the figure, the stroke factor increases as the cylinder becomes less rigid. Case 6, pivoted and rigidly guided is four times worse than fixed and rigidly guided.

After the basic length is found, (Figure 2-12) may be used. Thrust is plotted vs. basic length. The stop tube is found in the center of the graph and varies from 1 to 7 inches for a basic length of 35 to 100 inches.

Where thrust and basic length intersect, will determine the stop tube length.

BUCKLING

Cylinders may fail by buckling at a load less than the elastic limit of the material. Buckling is the sudden collapse of the rod at or above the critical load.

Failure of cylinders due to buckling is dependent on several factors:

1. The loading; ie. the resulting thrust forces on the rod.
2. The type of end conditions that exist on the

cylinder; ie. cylinder mounting style, external transverse forces on the cylinder.

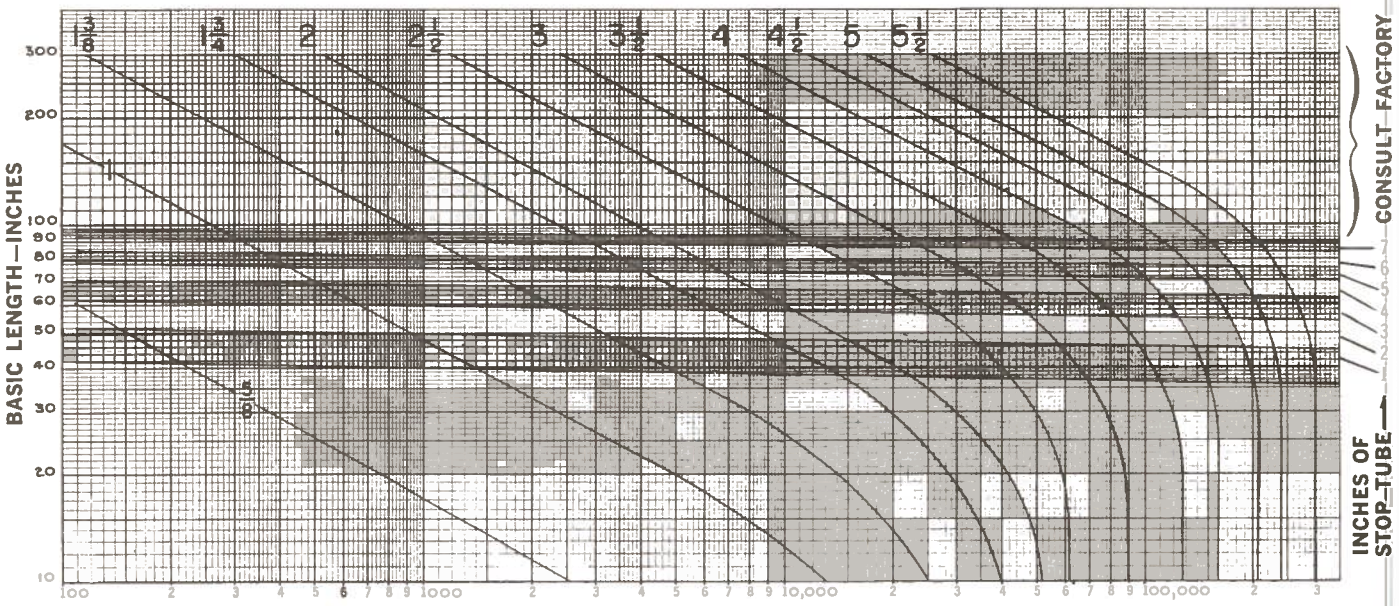
The type of end conditions on the cylinder was discussed in the previous section on selecting a stop tube.

SIZING A ROD TO PREVENT BUCKLING

After the basic stroke length of the cylinder is determined, (see selecting a stop tube) (Figure 2-12) is applied for the minimum rod size. The thrust and basic stroke length are intersected and the next largest rod is selected (rod line seen at a 45° angle left to right.)

ROD END CONNECTION	CASE	STROKE FACTOR
FIXED AND RIGIDLY GUIDED	I	.50
PIVOTED AND RIGIDLY GUIDED	II	.70
SUPPORTED BUT NOT RIGIDLY GUIDED	III	2.00
PIVOTED AND RIGIDLY GUIDED	IV	1.00
PIVOTED AND RIGIDLY GUIDED	V	1.50
PIVOTED AND RIGIDLY GUIDED	VI	2.00

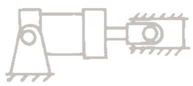
(Figure 2-11)



(Figure 2-12)

EXAMPLE: (2-6)

In a particular cylinder application, the designer developes 10,000# of thrust. You mention to him that for a stroke of 25 inches with the cylinder trunion mounted at the end and the load rigidly guided, (Figure 2-13) that he must have a stop tube and at least a 2 inch diameter rod. He says you are wrong. Prove who is correct.



(Figure 2-13)

SOLUTION:

To determine whether a stop tube is needed or buckling will take place, the basic stroke length of the cylinder is determined first. Consulting (Figure 2-14), we find that in this application we have a stroke factor of 2. The basic stroke length then becomes:

Basic Stroke Length = Stroke x Stroke Factor

Basic Stroke Length = 25 x 2 = 50 inches

With a basic stroke length of 50 inches and a thrust

of 10,000#, we can use (Figure 2-12) to find the stop tube length and the minimum rod size. (See Figure 2-15)

The stop tube length is 2 inches and is found at the right side of the graph. The rod diameter is 2 inches and is found by the first rod size moving vertically upward from the point of intersection. (2 inch rod selected because it is the next rod size after intersection point.)

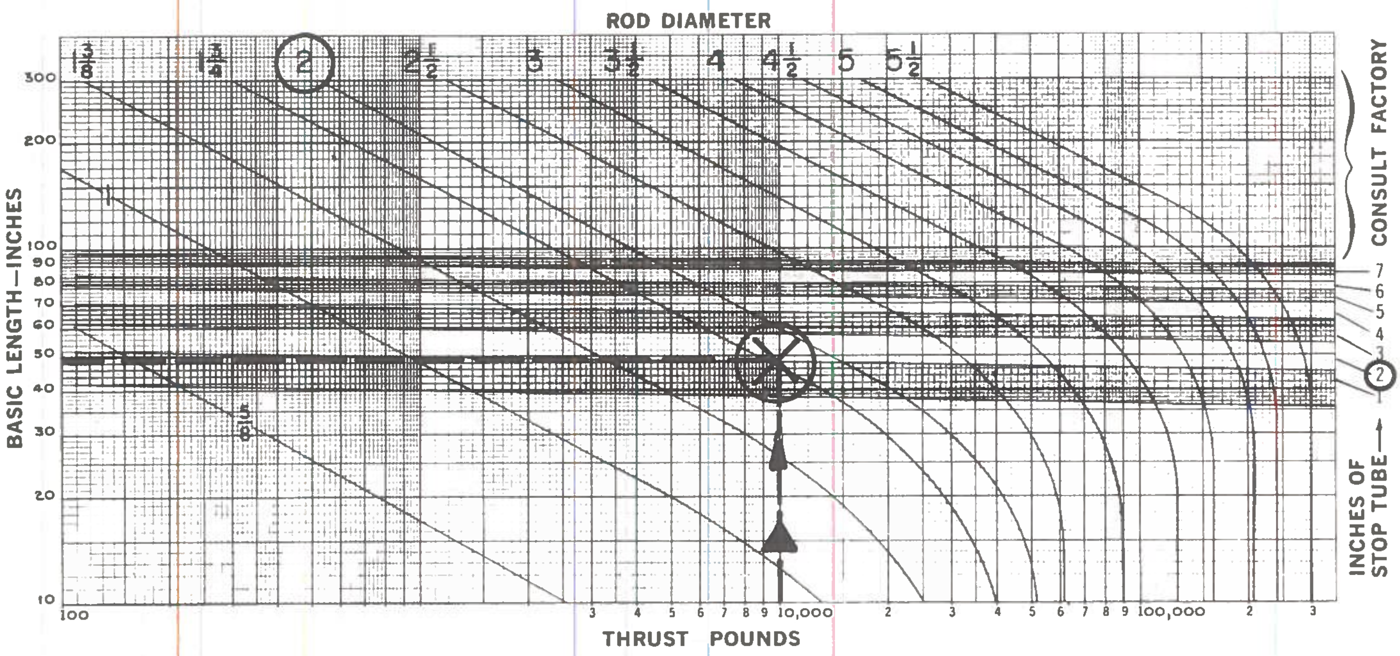
ROD END CONNECTION	CASE	STROKE FACTOR
FIXED AND RIGIDLY GUIDED	I	.50
PIVOTED AND RIGIDLY GUIDED	II	.70
SUPPORTED BUT NOT RIGIDLY GUIDED	III	2.00
PIVOTED AND RIGIDLY GUIDED	IV	1.00
PIVOTED AND RIGIDLY GUIDED	V	1.50
PIVOTED AND RIGIDLY GUIDED	VI	2.00

(Figure 2-14)

ACCELERATING A LOAD

In many applications, the acceleration of a cylinder load is not considered. To quantify this statement, we will say that if the load moves from zero to maximum velocity in cushion length, the acceleration can be considered negligible. Once the acceleration length starts to decrease, this additonal force becomes increasingly important. The underlying reason being that the force developed by a cylinder to a load is not only a function of the loading (friction & weight) but also a function of the mass of the load, the liquid to and from the cylinder, and the acceleration of both.

If one feels that acceleration need be considered, the following procedure will be helpful.



(Figure 2-15)

The thrust developed is a function of the force due to acceleration and friction:

Thrust = Fa + Fs
(2-14)

Where Fa = Force of Acceleration
Fs = Friction Force, Weight or other force that must be overcome.

The force due to friction is determined by the types of material which are sliding against one another and what type, if any, lubricant is used. Many times, this coefficient of friction can be found in a mechanical engineering handbook.

The equation will become:

$$Fs = us \times Wt. \quad (2\text{-}15)$$

Where Wt = Weight of object
us = Coefficient of friction

The force due to acceleration can be calculated from equation (2-16).

$$Fa = Me \times a \quad (2\text{-}16)$$

Where Me = Equivalent mass of liquid and weight
a = Acceleration

The equivalent mass is:

$$Me = \frac{\text{Wt load} + \text{Wt. Fluid}}{g} \quad (2\text{-}17)$$

Where g is force of gravity (32.2 ft/sec^2)
(9.8 m/s^2)

Wt load is the weight of the load and Wt fluid is the weight of the oil in the connecting lines between the valve and the cylinder.

The acceleration is defined as:

$$a(\text{ft/sec2}) = \frac{V\ (\text{in/m})}{t\ (\text{sec}) \times 720} \quad (2\text{-}18a)$$

OR

$$a(\text{m/s}^2) = \frac{V(\text{m/s})}{t\ (\text{s})} \quad (2\text{-}18b)$$

Where V = rod speed
t = time to go from 0 to the required rod speed (sec)
a = acceleration

Combining (2-14-15-16-17-18) we find:

$$\text{Thrust} = Fa + Fs = \frac{\text{Wt. load} + \text{Wt fluid}}{32.2} \times \frac{V}{t \times 720} + Fs$$

$$\text{Thrust}\quad (\text{N}) = \frac{\text{Wt load} + \text{Wt fluid}}{9.8} \times \frac{V(\text{m/s})}{t\ (\text{s})} + Fs$$

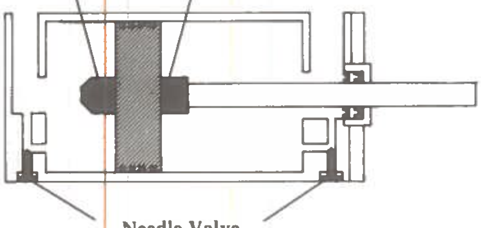
It is with this term "thrust" that chart (2-12) can be intersected to find the stop tube and minimum rod size. Also with thrust, one can determine the pressure needed to achieve the necessary accelera-

tion by letting thrust equal Fc or Ft in equations (2-4) & (2-5).

CUSHIONS

When hydraulic enrgy moving a cylinder's piston is suddenly stopped (as at the end of a cylinder's stroke), the liquid inertia is changed into a concussion known as "hydraulic shock". If a substantial amount of kinetic energy is stopped too abruptly, the resulting excess shock may damage the cylinder.

To protect against excessive shock, a cylinder can be equipped with cushions. Cushions slow down a cylinder's piston movement just before reaching the end of the stroke. Cushions can be applied at either or both ends of a cylinder.



(Figure 2-16)

A cushion consists of a needle valve flow control and a plug attached to the piston. The plug can be on the rod side, in which case it is called a cushion sleeve. Or, it can be on the cap end side, in which case it is called a cushion spear. (Figure 2-16)

Up until recently, the typical cushion sleeves and spears are for cushions were nothing more than tapered entrance portions. This produced a high initial pressure surge on high velocity applications.

However, new types of cushions are being introduced. The step cushion has been developed which cuts this initial pressure peak by 50%. All around performance of this cushion is superior to the old type of design.

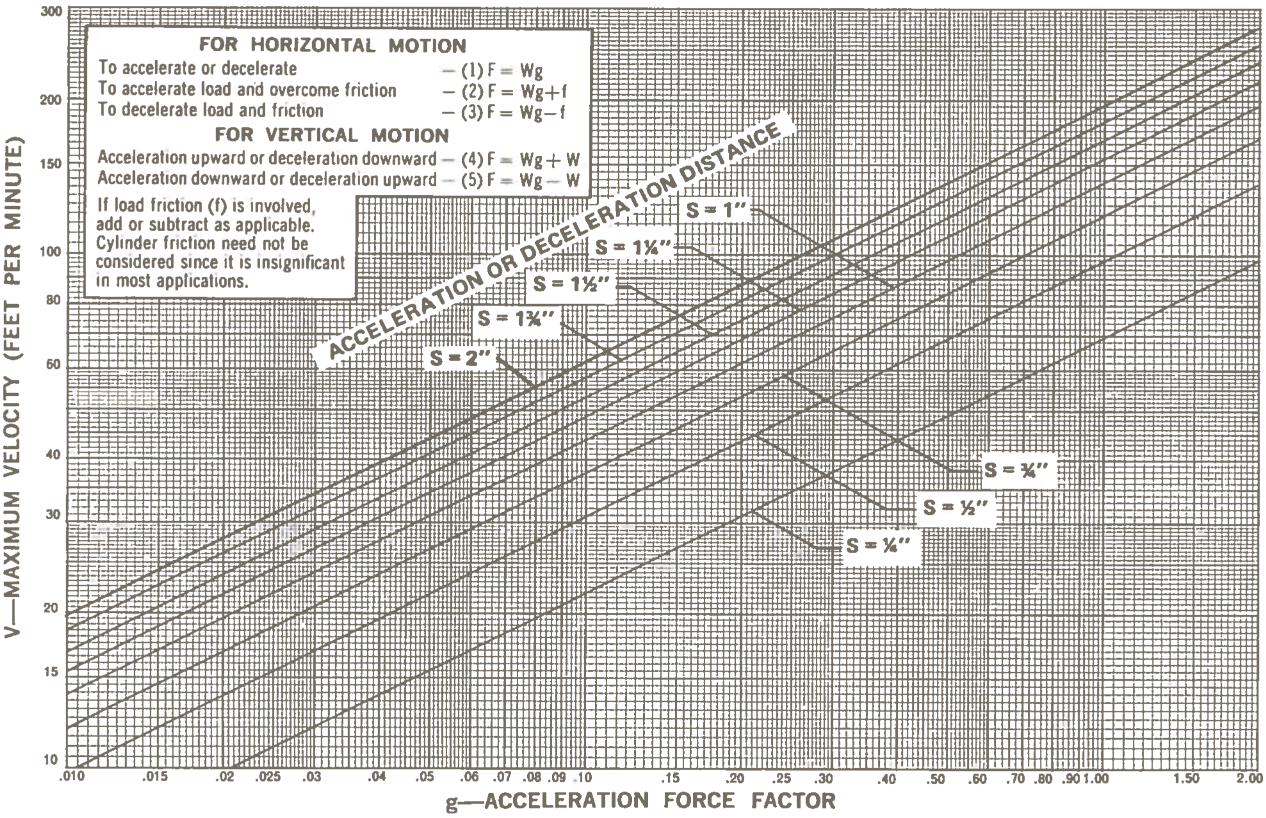
MAXIMUM DECELERATION WITH A CUSHION

When using a cushion to decelerate a moving load, the major consideration from the standpoint of cylinder selection is that the maximum pressure developed by the cushioning device does not exceed the pressure rating of the cylinder.

To determine expected cushion pressures on the cylinder, the following factors must be known:

1. total weight to be moved — fixed or variable
2. maximum piston speed
3. distance available for deceleration (cushion length)
4. direction of load (horizontal or vertical thrust or tension)
5. load friction: time cycle

After the above information is known, apply (Figure 2-17) to obtain the necessary acceleration. With this, the maximum pressures in the cylinder may be calculated.



(Figure 2-17)

To find the maximum pressure in a cylinder during deceleration, follow this procedure:

1. determine the cylinder-piston area Ap and annulus area Ae.
2. determine load orientation — horizontal or vertical
3. what is the load in pounds?
4. what is the friction in pounds?
5. what is the rod-speed in ft/min?
6. what is cushion length?
7. enter (figure 2-17) with rod speed and cushion length(s) and determine the acceleration factor.
8. apply the correct formula from (Figure 2-17) (upper left)
9. add the force developed by the relief valve setting on the non-cushioning area, to the value calculated in step 8.
10. divide the value calculated in step 9 by the piston or annulus area (depending on direction of motion) to find the maximum pressure. Compare it to non-shock rating to decide if it is safe.

EXAMPLE: (2-7)

A 4-inch bore, 10-inch stroke cylinder with a 2-inch diameter rod moves a weight of 20,000 pounds. It is mounted horizontally. The load has a coefficient of friction of .1. The non-shock rating of the cylinder is 5000 psi. Rod speed is 1200 in/min. Cushion length is 1 inch. What is the maximum pressure developed by the cushion? Relief Valve is set at 2000 psi.

SOLUTION: To solve the problem, we will employ the check list point by point.

1. Piston Area = Ap $= \frac{\pi 4^2}{4} = 12.56\ in^2$

   Annulus Area = Ae $= \frac{\pi}{4}(4^2 - 2^2) = 9.42\ in^2$

2. Horizontal load

3. Load is 20,000 lbs.

4. Friction = FS = Wt x us
   FS = 20,000 x .1
   FS = 2000 lbs. of friction

5. rod speed $= \frac{1200.0\ in}{min.} = \frac{100\ ft.}{min.}$

6. cushion length = 1 inch

7. enter (Figure 2-17) with V = 100 &
   S = cushion length = 1 inch

From the figure g = .52

8. applying the formula from the chart
   F = Wg - fr
   F = 20,000 (.52) - 2000
   F = 8400 lbs.

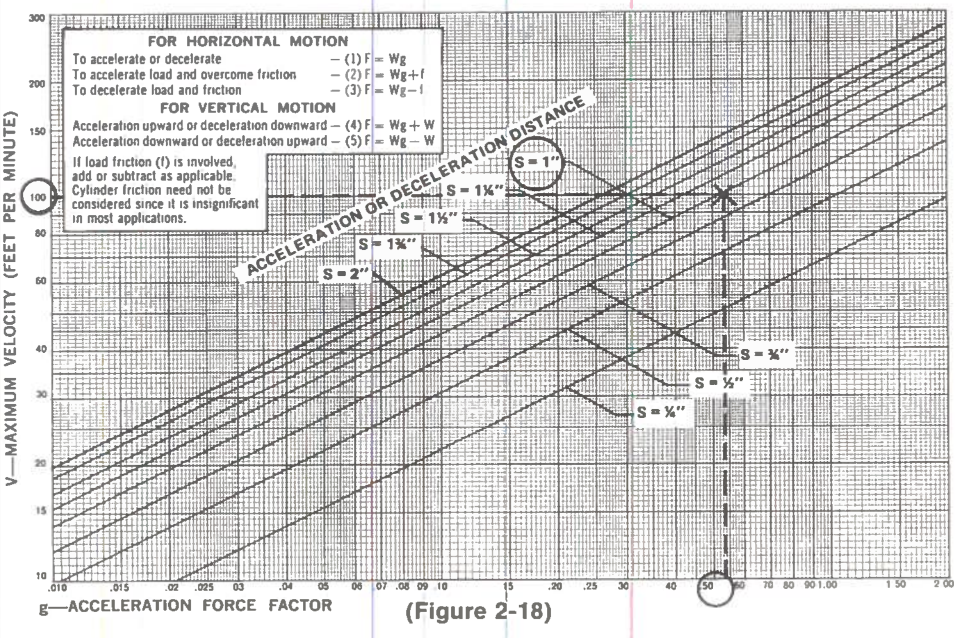
9. adding the additional force due to the pump
   F additional = Pc x Ap
   = 2000 x 12.56
   = 25,120 lbs.

   Adding to 8 we get —

   F total = 8400 + 25120 = 33520 lbs.

10. dividing 9 by the annulus area we have:
    Max pressure = 3558 psi

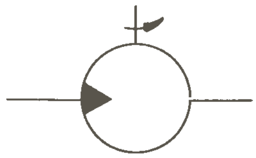
This value is less than the non-shock rating. Therefore, no problems should exist.

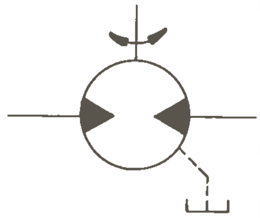


(Figure 2-18)

ROTARY ACTUATORS & MOTORS

Hydraulic motors or rotary actuators are similar to pumps which are supplied with a flow of high pressure fluid, which transmits power to a drive shaft. They accomplish work by letting high pressure oil act on the device's movable elements causing a drive to rotate. The efficiency of these devices is much more important, especially in the low RPM ranges and at startup. Guessing or rule-of-thumb is not sufficiently accurate. Graphs or charts should be consulted.


Hydraulic Motor Symbol (Uni-directional)


Bi-directional Motor Symbol

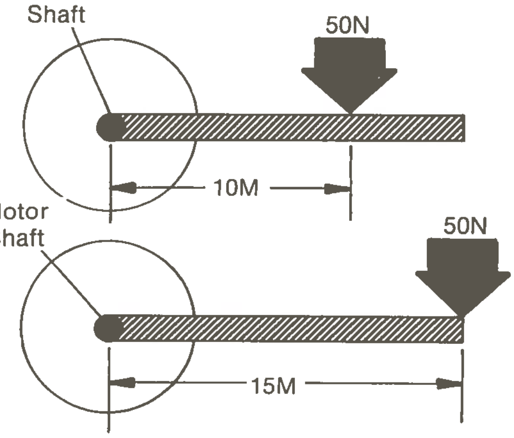
(Figure 2-19)

In rotary power two terms are quite common: RPM and torque. RPM or revolutions per minute, describe how rapidly the motor's shaft will rotate.

There are two types of torque: running torque and starting torque. The expression that often describes torque is:

Torque =	Force x	Distance from the center of the shaft
(lb.in)	(lb)	(in)
(N-M)	(N)	(M)

In the illustration, a force of 50 newtons is positioned on a bar which is attached to a motor shaft. The distance between the center of the shaft and the point of application of the force is 10 meters. This results in a torque or turning effort at the shaft of 500 newton meters (50 N x 10m) or 4429 pound inches. (Figure 2-20)



(Figure 2-20)

If the 50 newtons were located 15 meters along the bar, the turning effort generated at the shaft would equal to a load of 750 N-M located one meter from the center of the shaft.

From these examples, we can see that the farther the given force is from the shaft, the larger the resultant torque at the shaft. It will also be noted that torque does not necessarily involve any movement.

A resisting object attached to a motor shaft results in a torque as described above. This, of course, is a resistance for the motor which must be overcome by hydraulic pressure acting on the motor's rotating group.

Hydraulic motors operate as a result of a pressure imbalance on the appropriate members of the rotating group which results in the rotation of the shaft. This imbalance is generated in different ways depending on the motor type.

Running torque is described as the torque developed by a motor while the shaft is in motion. Starting torque is that torque which the motor can deliver as motion of the shaft starts. For hydraulic motors, starting torque varies. It is typically less than the running torque.

Mechanical power from a motor can be calculated.

The formula is:

$$\text{Output HP} = \frac{\text{RPM x Torque}}{63025}$$

(2-21a)

Where RPM is in revolution/minute
Torque is in inch-pounds

$$\text{Output power (K Watts)} = \frac{\text{Speed (rad/sec) x torque (N-M)}}{1000}$$

Equation (2-21) tells us the amount of power which can be obtained from any motor at a particular speed and torque. To drive a hydraulic motor we must input a particular flow rate and pressure. As you recall from the section on cylinders, the hydraulic power (equation 2-7a) equaled the mechanical power. (equation 2-8). This is because for the sake of complicity, the cylinder was assumed to be 100% efficient. In this case, when we supply the hydraulic horsepower equation (2-7), it will not equal equation (2-21), (2-7) will have a higher value. If the value obtained in (2-21) is divided by (2-7) the efficiency of the motor is obtained.

$$\text{Efficiency} = \frac{\text{HP out}}{\text{HP in}} = \frac{\text{Equation (2-21)}}{\text{Equation (2-7)}} \text{ x } 100$$

(2-21)

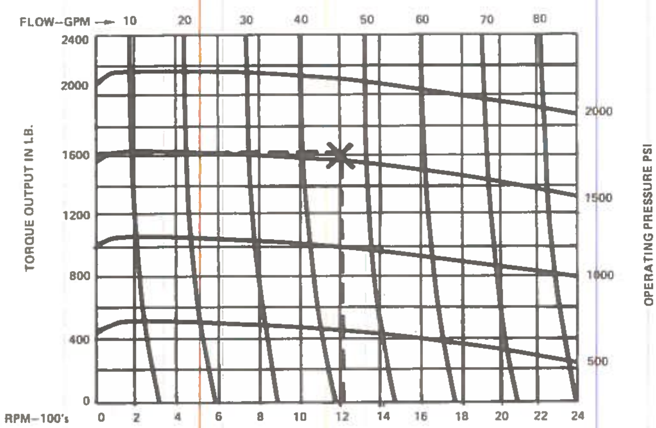
$$\text{Motor Efficiency} = \frac{\text{RPM x Torque (inch-pounds)}}{\text{GPM x PSI x 36.7}} \text{ x } 100$$

(2-22a)

$$\text{Motor Efficiency} = \left(\frac{\text{(rad/sec) x Torque (n-m)}}{\frac{\text{Flow rate (LPM) x Pressure (bar)}}{600}} \right) \text{ x } 100$$

(2-22b)

The horsepower lost due to the inefficiency of the motor is converted into heat. This heat causes a rise in the temperature of the oil. If the system's oil temperature gets too high, a heat exchanger must be employed.



(Figure 2-21)

EXAMPLE (2-8)

A certain motor exists in our system. The motor has a "Q" series marked on the nameplate. We know the motor runs at 1200 RPM's and develops 1600 inch-pounds of torque. What is:

1. the flow rate?
2. the pressure needed to operate the motor?
3. the starting torque?
4. the efficiency?
5. the heat generated?

SOLUTION:

The first thing that must be done is find the performance data of this motor. In the catalog the series motor is found and a performance chart located. (Figure 2-21)

On this chart, we locate the speed (RPM) along the bottom at 1200. This is a vertical line. The torque output is located at the left on a horizontal line. Let the two lines intersect. It is where these two lines intersect that the flow rate (somewhat vertical line curving to the right) and the pressure (somewhat horizontal lines curving down) can be found. In this case, a line drawn parallel to the 40 GPM line approximately 46 GPM and 1500 PSI line directly intersects the "x". Therefore, this motor must be supplied with:

a. 46 GPM
b. 1500 PSI

The starting torque is given as a function of running torque. In this case, the catalog says it is 95% of the running value. Therefore:

Starting torque = .95 x 1600 in.lb. = 1520 in.lb.

The motor efficiency can be obtained by using formula (2-22a)

$$\text{Motor Efficiency} = \frac{\text{RPM x Torque}}{\text{GPM x PSI x 36.7}} \text{ x } 100$$

Where RPM = 1200
torque = 1600 in.lbs.
GPM = 46
PSI = 1500 psi

$$\text{Efficiency} = \frac{(1200) \text{ x } (1600) \text{ x } 100}{(46) \text{ x } (1500) \text{ x } 36.7} = 75.8\% \approx 76\%$$

The heat generated deals with the amount of wasted horsepower. The wasted horsepower, in this case, is 24% of the input horsepower. The input horsepower can be obtained from education (2-7) which reads:

Input HP = GPM x PSI x .000583
Input HP = 40.22 HP
Horsepower loss = 40.22 x 24% = 9.65 HP

This is the amount of horsepower that is turned into heat. It may be converted into BTU's/hr by multiplying by 2547. Therefore, we have:

Heat generated = 9.65 x 2547 = 24,590 BTU/hr

This means that if the temperature of the oil is to remain constant, 24,590 BTU's/hr of heat must be dissipated by radiation, convection or conduction.

If the machine itself cannot dissipate all this heat, a heat exchanger must be added. The total heat generated in a circuit will be considered later.

The accuracy of each type is different. This accuracy will be covered in the chapter on flow controls.

CONCLUSION

In this chapter, we have covered the final converting element in a fluid power system. This element takes on the form of cylinders, rotary acutators or motors.

When we approach cylinder calculations, the following equations will be used:

$$F \text{ (pounds force)} = P \text{ (PSI)} \times A \text{ (in}^2\text{)}$$

$$Ap \text{ (in}^2\text{)} = \text{area of piston} = \frac{\pi dp^2 \text{ (in)}}{4}$$

$$Ae \text{ (in}^2\text{)} = \text{Annulus Area} = (Ap - Ar) = \frac{\pi (dp^2 - dr^2)}{4} \text{(in)}$$

$$Fc \text{ (pounds force)} = Pc\text{(psi)} \times Ap\text{(in}^2\text{)} - Pr\text{(psi)} \times Ae\text{(in}^2\text{)}$$

$$Fc\text{(N)} = (Pc\text{(bar)} \times Ap\text{(cm}^2\text{)}) - Pr\text{(bar)} \times Ae\text{(cm}^2\text{)}/0.1$$

$$Ft\text{(pounds force)} = Pr\text{(psi)} \times Ae\text{(in}^2\text{)} - Pc\text{(psi)} \times Ap\text{(in}^2\text{)}$$

$$Ft\text{(N)} = (Pr\text{(bar)} \times Ae\text{(cm2)}) - Pc\text{(bar)} \times Ap\text{(cm}^2\text{)}/0.1$$

$$V\text{(inches/min.)} = \frac{Q \text{ (GPM)} \times 231}{A \text{ (in}^2\text{)}}$$

$$V\text{(meters/second) (M/S)} = \text{Flow (LPM)} / (A\text{(cm}^2\text{)} \times 6)$$

$$\text{Input Power(kilowatts KW)} = (\text{flow(LPM)} \times \text{Pressure(bar)})/600$$

$$Hp \text{ Input} = GPM \times PSI \times .000583$$

$$\text{Output Power (KW)} = \frac{\text{Speed (m/s)} \times \text{force (N)}}{1000}$$

$$Hp \text{ out} = V \text{ (in/min)} \times \frac{F \text{ (pounds force)}}{396{,}000}$$

For regenerative circuits:

$$Fc\text{(pounds force)} = Ar\text{(rod area)} \times P \text{ (PSI)}$$

$$Fc\text{(N)} = (Ar\text{(cm}^2\text{)} \times P\text{(bar)}) / 0.1$$

$$\text{Rod Area (in}^2\text{)} = \pi dr^2/4 = .7854 \, dr^2 \text{ (in)}$$

$$Ar\text{(cm}^2\text{)} = 0.007854 \, dr^2 \text{ (mm)}$$

$$\text{Rod Velocity (extending)} = V \text{ (in/min)} = \frac{Q \text{ (GPM)} \times 231}{Ar \text{ (in}^2\text{)}}$$

$$V(m/s) = \frac{Q\ (LPM)}{Ar(cm^2) \times 6.0}$$

Basic Stroke Length (Stroke) = Stroke x Stroke Factor

$$Thrust = \frac{Wt.\ load + Wt.\ fluid}{32.2} \times \frac{V\left(\frac{in}{min}\right)}{t(sec) \times 720} + Fs$$

$$Thrust\ (N) = \left[\frac{(Wt.load + Wt.fluid)}{9.8} \times \frac{V(m/s)}{(s)}\right] + Fs$$

For hydraulic motors, a new set of equations were developed.

Torque (# in) = Force(pounds-force) x distance (in)

Torque (N-m) = Force (N) x Distance (m)

$$Output\ HP = \frac{RPM \times Torque\ (inch\text{-}pounds)}{63025}$$

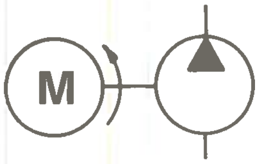
Also, the efficiency of a hydraulic motor is:

$$Output\ power\ (KW) = \frac{Speed(rad/s) \times Torque(n\text{-}m)}{1000}$$

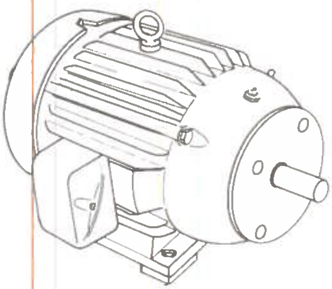
$$Motor\ efficiency = \frac{(Rad/sec) \times Torque\ (n\text{-}m)}{GPM \times PSI \times 36.7} \times 100$$

$$Motor\ efficiency = \frac{(rad/sec) \times torque\ (n\text{-}m) \times 100}{\frac{(flowrate\ (LPM) \times Pressure(bar)}{600}}$$

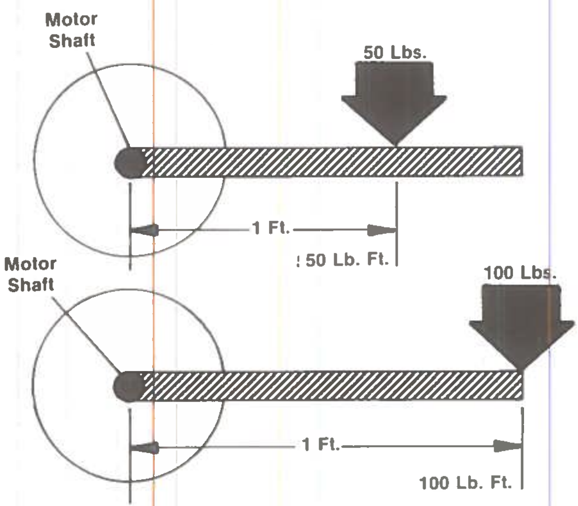
CHAPTER 3
THE ELECTRIC MOTOR



(Figure 3-1)



(Figure 3-2)



(Figure 3-3)

In a hydraulic system, energy is transmitted under the form of a moving pressurized liquid: this is hydraulic power. Generation of hydraulic power is the responsibility of a positive displacement pump and its prime mover which in an industrial system is an electric motor. (Figure 3-1)

ELECTRIC MOTORS

Industrial hydraulic systems are a means of energy transmission. They are dependent on a source of input power which is usually electrical. Electric power as it is delivered to a system under the form of voltage and current, is transformed into rotary mechanical power by an electric motor and passed on to a pump. (Figure 3-2)

ROTARY MECHANICAL POWER

Rotary mechanical power developed by an electric motor can be divided into RPM and torque.

Torque is a rotary or turning effort expressed in units of force and distance. One unit for measuring torque is the pound-foot or newton meter.

Torque indicates where a force is in relation to the motor shaft and is described as:

Torque = Force x Distance from Shaft

(3-1)

In the illustration, a 50 lb. (21.5 N) force is positioned on a bar which is attached to a motor shaft. Distance between shaft and force is 1 ft. (25.4 mm) resulting in a turning effort at the shaft of 50 lb.ft. (6.55 N-M). If 100 lbs. were located 1 ft. along the bar, turning effort generated at the would be 100 lb. ft. (13.1 N-M) (Figure 3-3)

In a hydraulic system, a pump connected to an electric motor shaft is a resistive torque just as the above examples. Before the rotating elements of a pump will turn, its resistance must be overcome by the torque developed by an electric motor. The higher the pressure a pump must develop to push out its flow, a greater is the resistive torque for an electric motor.

Torque does not involve movement; it indicates the amount of turning effort in terms of force at a distance. If torque is incorporated with speed (RPM); the result is rotary work being performed at a certain rate which is mechanical horsepower.

$$\text{Motor horsepower} = \frac{\text{RPM x Torque (lb.ft.)}}{5252}$$

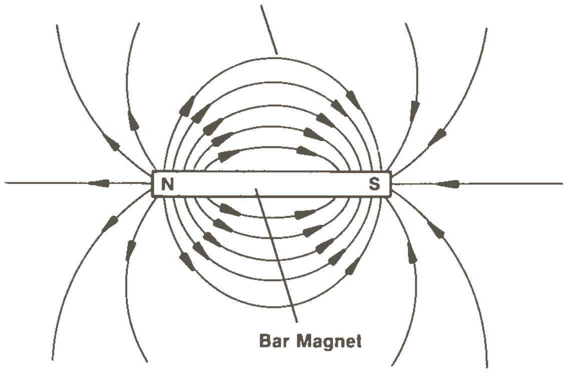
(3-2a)

$$\text{Motor power (KW)} = \frac{\text{RPM x Torque (N-M)}}{7043}$$

(3-2b)

MAGNETIC FIELD

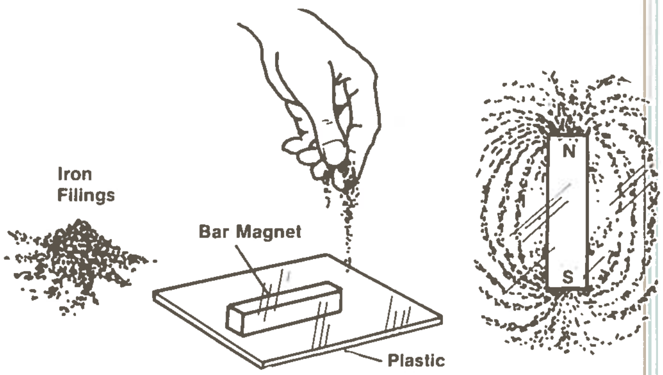
An electric motor transforms electric power into torque and RPM by developing an alternating magnetic field which forces a mechanical element to move. This action can be demonstrated with ordinary magnets. (Figure 3-4)



(Figure 3-4)

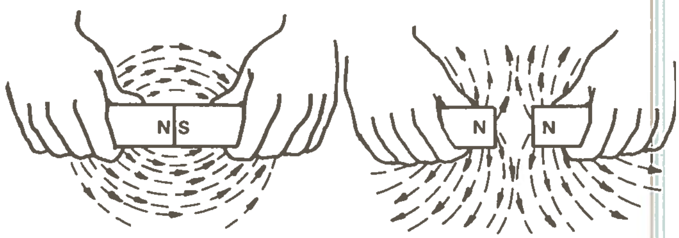
We know from experience that magnets exert attractive forces on iron and ordinary steel objects. This can be seen when a magnet is brought near a pile of tacks or even in a scrap yard where huge magnets attached to cranes move rusty metal from place to place. This attraction is the result of a magnetic field surrounding a magnet.

Every magnet has a magnetic field made up of lines of force which exit at its north (+) pole and enter at its south (-) pole. Magnetic lines of force are invisible, but their effect can be seen by sprinkling iron filings on clear plastic under which a magnet is located. The filings will take the shape of the magnetic field. (Figure 3-5)



(Figure 3-5)

Anyone who has played with two bar magnets knows that in one position magnetic poles of each magnet are attracted. When rearranged, magnetic poles may be repulsed or driven away. Attraction occurs between unsimilar poles (north + south -); repulsion exists between similar poles. (Figure 3-6)

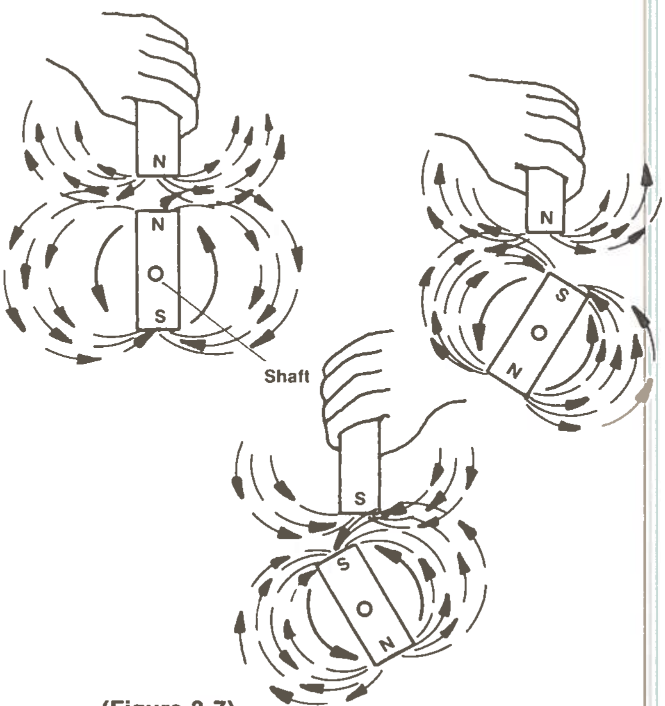


(Figure 3-6)

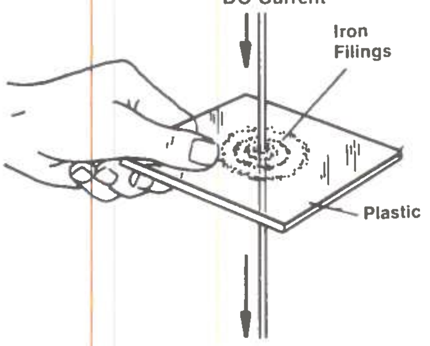
This basic phenomenon is caused by each magnet's inherent magnetic field. Depending on the field strength, it is very difficult, if not impossible, to get similar poles to make contact.

MAGNET MOTOR

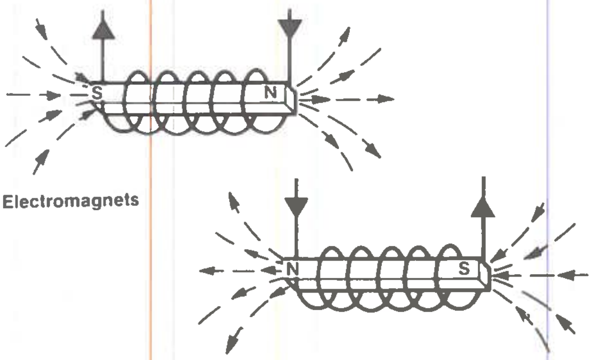
If a small shaft or rod supported a bar magnet through its center, (Figure 3-7), another magnet could cause the unit to rotate. By placing the positive end of the free magnet near the positive end of the shaft-supported magnet, the shaft magnet would be repulsed or driven away causing rotation. As the positive end moved farther away, the negative end moving closer would feel the attractive force of the static positive pole. If then, the static magnet were turned at the proper time so that its negative pole repulsed the oncoming negative pole, movement would continue. As long as the magnetic field kept changing or alternating at the proper time, shaft and magnet would turn indefinately. This describes the operation of a bar-magnet motor. An electric motor also operates by means of an alternate magnetic field, but its field is generated by an electromagnet. (Figure 3-8)



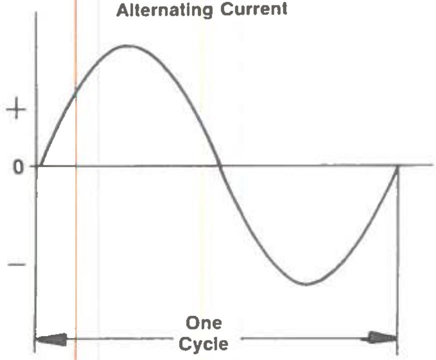
(Figure 3-7)



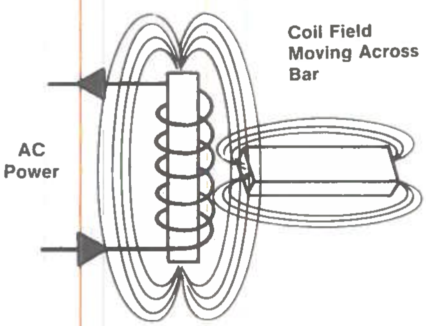
(Figure 3-8)



(Figure 3-9)



(Figure 3-10)



(Figure 3-11)

ELECTROMAGNET

As an electric current passes through a wire, a magnetic field sets up around the wire. This effect can be seen by sprinkling iron filings on a clear plastic sheet through which a wire is located. Connecting the wire across the terminals of a battery, we find that an electric current will cause filings to take the shape of its magnetic field.

If copper wire carrying direct current from a battery were wound around an unmagnetized iron bar, the magnetic field would be stronger and the iron bar would turn into a magnet with respective positive and negative poles. By changing the direction of direct current flow through the wire, positive and negative poles would reverse.

The coiled wire and iron bar assembly is known as an electromagnet. (Figure 3-9)

ALTERNATING CURRENT

The usual source of industrial electric power has an alternating current (AC). AC electricity can be transmitted efficiently from generating stations, over long distances to industrial plants and to an electric motor of a hydraulic system. (Figure 3-10)

Plotting alternating current on a graph, we could see that as time increased, the current varied between positive and negative peak values. As current moves from a zero point on the graph to positive peak, down to negative peak and back to zero, one cycle is completed. In the United States, AC electricity is usually 60 cycles per second or 60 hertz (Hz).

INDUCTION

In electrical systems, as a changing magnetic field cuts across a conductive material, an electric force is generated in the conductor. This pnenomenon is known as induction. (Figure 3-11)

Alternating current pulsing through an electromagnet coil at 60 Hz generates an alternating magnetic field which changes from positive to negative 60 times per second. As this change occurs, magnetic lines of force expand and collapse. If they are allowed to cut across a metal object, an electric force will be generated in the object.

For example, an unmagnetized, ordinary steel bar placed next to an electromagnet carrying alternating current, would be cut by expanding and collapsing lines of magnetic force this results in a magnetic field setting up in the bar, but this magnetic field is opposite in charge to the electromagnet. As the electromagnetic field of the coil

switched from positive to negative, the bar magnetic field would change from negative to positive. The magnetic field of the bar has been induced by, but opposes the alternating field of the electromagnet. This action is seen in an induction motor.

3-PHASE SQUIRREL CAGE INDUCTION MOTOR

Because of its simplicity and low cost, a common electric motor for an industrial hydraulic system is a 3-phase, squirrel cage, induction motor which consists of basically two elements — stator and rotor (Figure 3-12)

STATOR

A stator is the part of an electric motor which remains stationary. It is made up of electromagnets which develop a rotating magnetic field around the rotor. A motor stator is sometimes referred to as a "field". (Figure 3-13)

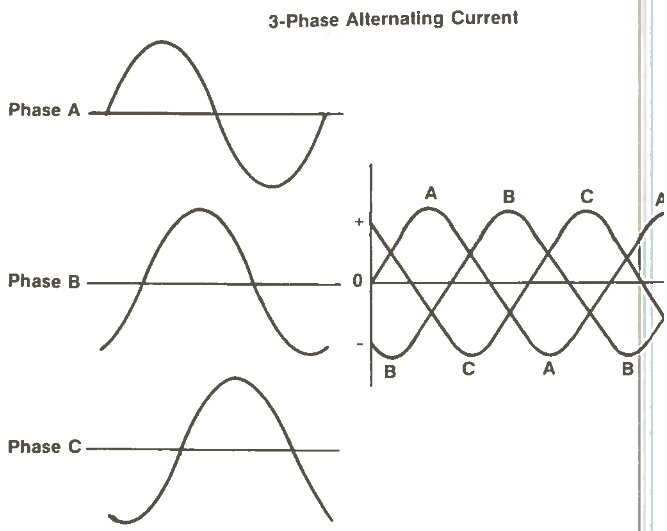
In the diagram, three iron cores are wound with wire and positioned equally about the center of a circle. The coils are connected to a 3-phase source of power. In a 3-phase power supply, three alternating currents from a single source are available to the motor; but, each current cycles differently so that they reach peak values at different times. Our 3-phase motor takes advantage of this face by connecting individual coils to one alternating current of a 3-phase supply. This means a fraction of a second later coil B will have its maximum field, then coil C. The magnetic field rotates about the center of the stator.

To develop an even stronger magnetic field, additional electromagnetic coils are used. In the illustration of a two pole stator, a common iron core is used from one pole. Many industrial motors are equipped with either four or six poles. (Figure 3-14)

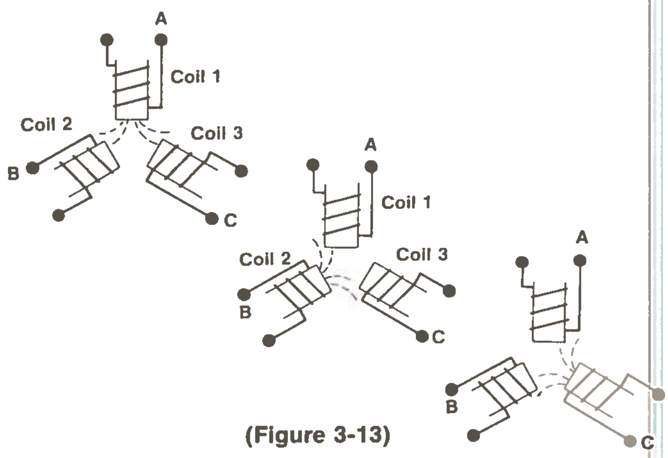
ROTOR

A rotor or armature of an electric motor is the mechanical element which rotates. It is positioned in the center of the stator windings and is supported by the motor's output shaft which is an integral part of the assembly.

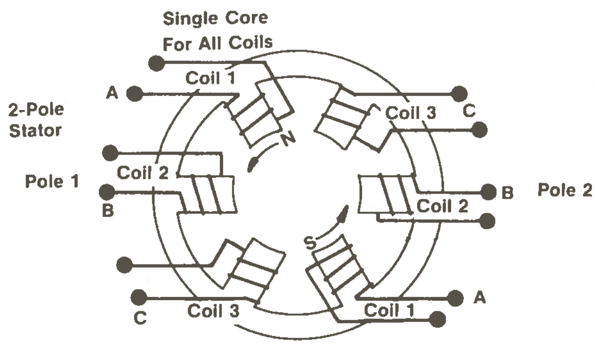
The rotor of a squirrel cage motor is essentially made up of copper or aluminum bars which are connected together by rings at either end. This gives the appearance of a squirrel cage, hence the name. (Figure 3-15) However, the rotor of a modern motor of this type will appear as a smooth cylinder since a layer of material covers the bars.



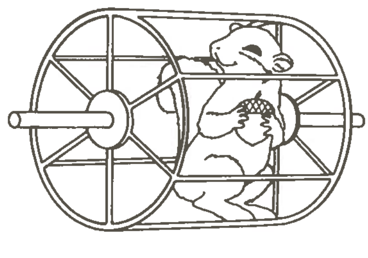
(Figure 3-12)



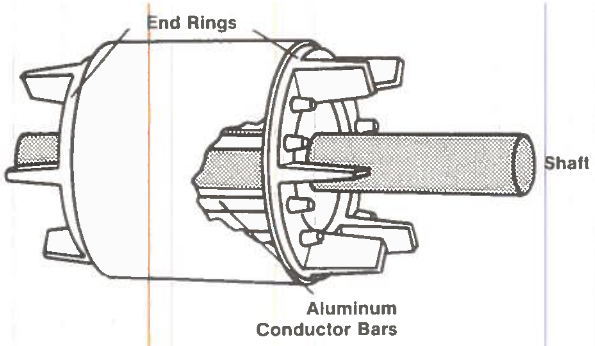
(Figure 3-13)



(Figure 3-14)

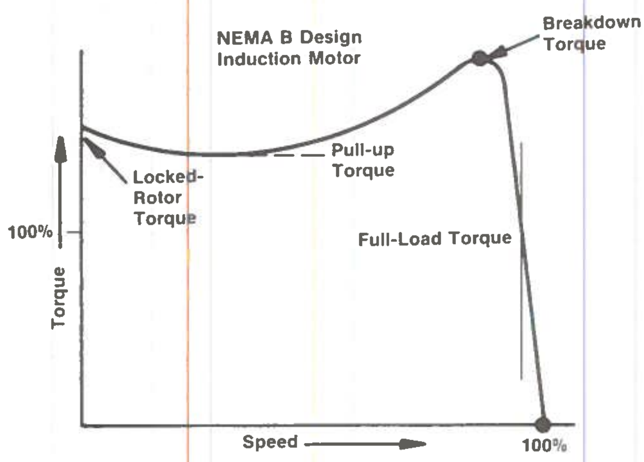


(Figure 15a)



Squirrel Cage Rotor

(Figure 15b)



(Figure 3-16)

Since it is an induction motor, rotation of a squirrel cage rotor is dependent on an induced electromagnetic field set up by the stator. Strength of the rotor field is influenced by the amount of current in the stator coils. As a motor load becomes greater, as in the case of a pump attached to its shaft experiencing more resistance to flow, current in stator coils increases so that rotor field will increase keeping the shaft turning at the same point.

NEMA MOTOR DESIGN

All 3-phase, squirrel cage induction motors are not the same. Depending on the design of the rotor, the induced magnetic field and operating characteristics will vary. For the sake of standardization, members of the National Electrical Manufacturers Association (NEMA) working with industrial users decided that four designs would satisfy industrial needs. These designs are labelled A, B, or C or D with the design B being seen most frequently in industrial systems.

The actual class definitions are the following:
Nema Class A — A general purpose motor with starting currents 5 to 7 times rated and starting torque 150% rates.
Nema Class B — A general purpose motor with starting current 4½ to 5 times rated, starting torque 150% rated.
Nema Class C — A double squirrel cage motor with high starting torque characteristics. Starting current 4½ to 5 times rates, torque 225% rated.
Nema Class D — High torque, high resistance motor with starting current 4 to 4½ times rated and starting torque 275% rating.

There are class E and F motors also. However, these are little used since nearly all applications are usually met better by the other classes.

NEMA TORQUE RATING

As was shown previously, torque indicates turning effort. An electric motor has different values of torque over its speed range.

A speed-torque performance curve for a NEMA B design motor is illustrated. This curve points out a motor's locked-rotor torque, pull up torque, and breakdown torque. (Figure 3-16)

Locked-rotor torque is starting torque; it indicates a particular motor's minimum capability of developing torque from an at rest position or while stalled.

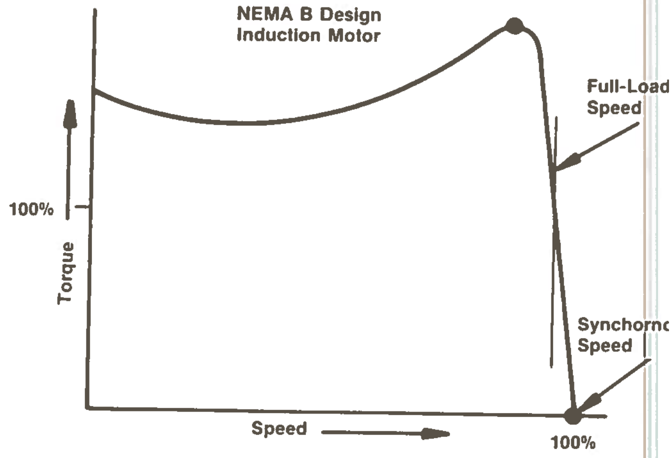
Pull-up torque is the minimum torque which a motor can develop while accelerating its load from zero to a point where breakdown torque occurs.

Breakdown troque is the maximum torque a motor will develop while not significantly changing its speed and while working at rated voltage and frequency.

Full-load torque is the torque required to develop rated motor horsepower at full speed.

NEMA SPEED RATING

A speed-torque performance curve for a NEMA B design motor points out a motor's synchronous and full-load speeds. (Figure 3-17a)



(Figure 17a)

Speed (RPM) of a squirrel cage motor depends on line frequency and the number of poles in the motor stator. With 3-phase power and a line frequency of 60 Hz (cycles per second), squirrel cage motors with four poles operate at 1800 RPM and motors with six poles run at 1200 RPM.

Speeds of 1200 RPM for a six pole motor and 1800 RPM for a four pole motor are known as synchronous speeds which means the rotor is turning at the same pace as the rotating magnetic field in the stator. In actuality this does not occur, the rotor turns slower than synchronous speed. This action allows the rotor to cut across magnetic lines of force of the stator resulting in additional current and an increased magnetic field at the rotor.

A motor turning at slightly less than synchronous speed under no-load condition, will deviate from synchronous speed to a greater extent as load increases. The difference between synchronous speed and the speed of the motor as it is loaded is known as "slip."

Slip of a squirrel cage motor is generally less than 5% at full load. It is common to refer to these as constant speed motors since at full load, speed varies little.

On their nameplates, many manufacturers indicate shaft speed at full load and not synchronous speed. For example, a motor with a synchronous speed of 1800 RPM may have a speed of 1750 RPM at full load which would be indicated on the nameplate.

Full load speed for various horsepower motors can be seen from the illustrated chart. (Figure 3-17b)

Synchronous Vs. Full-Load RPM

HP	Synchronous RPM	Full-Load RPM
½	900	850
¾	1,200	1,140
	900	850
1	1,800	1,735
	1,200	1,140
	900	870
1½	3,600	3,475
	1,800	1,715
	1,200	1,160
	900	865
2	3,600	3,495
	1,800	1,725
	1,200	1,155
	900	865
3	3,600	3,510
	1,800	1,745
	1,200	1,160
	900	865
5	3,600	3,510
	1,800	1,745
	1,200	1,150
	900	860
7½	3,600	3,530
	1,800	1,755
	1,200	1,155
	900	860
10	3,600	3,520
	1,800	1,750
	1,200	1,155
	900	870
15	3,600	3,520
	1,800	1,750
	1,200	1,170
	900	870
20	3,600	3,520
	1,800	1,750
	1,200	1,170
	900	875

HP	Synchronous RPM	Full-Load RPM
25	3,600	3,525
	1,800	1,765
	1,200	1,170
	900	875
30	3,600	3,530
	1,800	1,750
	1,200	1,165
	900	875
40	3,600	3,530
	1,800	1,755
	1,200	1,170
	900	875
50	3,600	3,545
	1,800	1,760
	1,200	1,170
	900	875
60	3,600	3,545
	1,800	1,770
	1,200	1,175
	900	875
75	3,600	3,555
	1,800	1,770
	1,200	1,175
	900	880
100	3,600	3,550
	1,800	1,775
	1,200	1,175
	900	880
125	3,600	3,560
	1,800	1,780
	1,200	1,175
150	3,600	3,560
	1,800	1,770
200	3,600	3,560
	1,800	1,770
250	3,600	3,560

(Figure 17b)

To calculate the no load speed of an induction motor, the following formula may be used:

$$N\ (rpm) = \frac{120\ F\ (Hz)}{P}$$

(3-3)

Where F is the line frequency in Hertz
P is the number of poles

The answer we obtain from (3-3) will be the speed of the motor in RPM with no-load. When the motor begins to load the speed of the motor will decrease;

EXAMPLE: (3-1)

A four pole induction motor is being run in Boston, Mass. with standard 3-phase voltage. What is its speed?

Employing (3-3) we have:

$N = 120\ F/P$

Since standard voltage is used, F = 60 Hz.

The number of poles is given and P = 4.

$$N = \frac{120\ (60)}{4} = 1800\ RPM$$

Under no load conditions.

MOTOR ENCLOSURE TYPE

A motor is enclosed to protect its vital internal parts from damage due to its environment. While others protect a potentially explosive atmosphere from igniting by enclosing the motor to prevent ignition. The different types are discussed below.

Motor types are divided into two classes — open or totally enclosed.

OPEN — The open motor is one having ventilating openings for free air cooling.

DRIP-PROOF — The drip-proof motor is constructed so that drops of liquid falling on the motor at an angle of 15% or less from vertical cannot enter. This is intended for indoor location, where the air is fairly clean.

GUARDED — Ventilated openings are limited to specific size and shape to prevent insertion of fingers or rods to avoid accidental contact with rotating or electrical parts.

SPLASH PROOF — Ventilation openings are constructed so that drops or solids falling on the motor at an angle less than 100° from the vertical, will not enter this motor. It is intended for more severe service than the drip proof. If better protection is needed, the trend is toward drip proof with sealed insulation or a totally enclosed motor.

WEATHER PROTECTED TYPE 1 — An open motor designed to minimize the entrance of rain, snow or other particles and prevent the passage of a

cylindrical rod of ¾" diameter. The housing is similar to drip proof, with the addition of outdoor type bearings, sealed insulation and screens on ventilation openings.

WEATHER PROTECTED TYPE 2 — In addition to the above, the air flow entering the motor is baffled to prevent airborn particles from contacting the electrical media.

TOTALLY ENCLOSED — This type of enclosure prevents the free exchange of air between the inside and outside of the case but is not airtight.

TOTALLY ENCLOSED NONVENTILATED (TENV) — Totally enclosed motor with no means of cooling the enclosed parts. It is restricted to small motor sizes.

TOTALLY ENCLOSED FAN COOLED (TEFC) — Motors equipped with an exterior fan internal to the motor, external to the enclosed part. The external fan spreads a blanket of air on the totally enclosed motor frame thereby dissipating heat. This enclosure is popular in dusty, dirty and corrosive atmospheres.

ENCAPSULATED — An encapsulated motor is an open motor whose windings are covered with a heavy coating of material to protect them from moisture, dirt and abrasion. If the coils are totally encapsulated, this motor can be used in applications where only totally enclosed motors were employed.

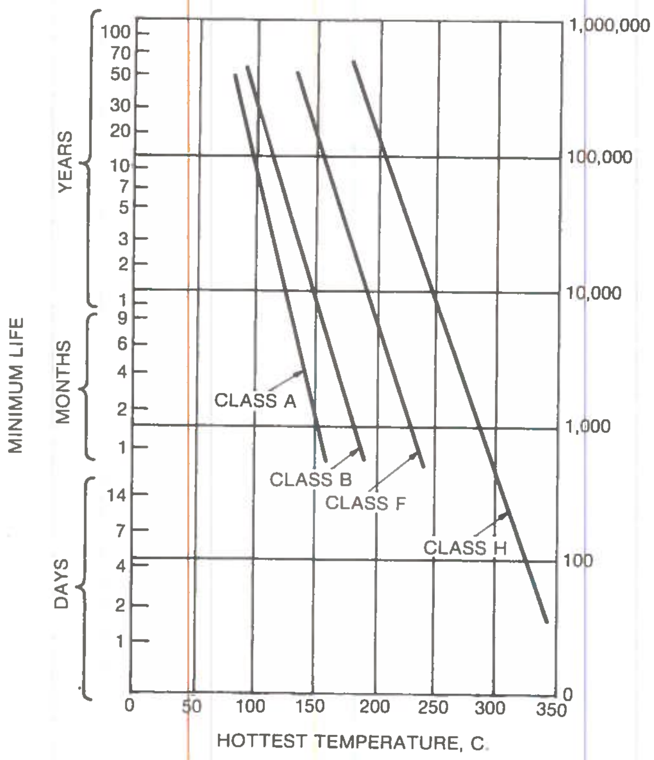
EXPLOSION PROOF — This motor is designed to prevent ignition of gas or vapor surrounding the motor. It is evident that the motor is impervious to the elements surrounding it.

NEMA FRAME SIZE

A standard system has been adopted for frame sizes of motors up to 500 HP at 1800 RPM. This is followed by most companies manufacturing motors. This frame number standardizes certain motor mounting dimensions.

Frame numbers are not arbitrarily chosen. Their significance can be found in NEMA standard MGH101.

Previously, frame sizes were designated by letters. The two most common were "U" and "T" frame motors. A "U" frame motor consisted of a heavier frame than the comparable "T". With this heavy "U" frame, the motor could be intermittenly over-loaded longer because of its heavy mass (meaning higher thermal capacity). In this way, the motor would not overheat. This type is still used for heavy duty applications such as in steel mills and automotive plants.



(Figure 3-18)

MAXIMUM OUTPUT

The maximum output of a machine, as discussed in the previous section, is determined by the heat generated within the motor in a particular period of time. If the temperature is too high or the time period is too long, the insulation may begin to breakdown. Breakdown in insulation is directly related to the life expectancy of the motor. When insulation begins to deteriorate, the following phenomenon takes place:

1. Oxidation and brittle hardening
2. Loss in durability

A rough idea of the life of insulation is that the time to failure of an organic insulation is halved for each 14 to 18°F temperature rise. A graph is shown. Short, infrequent periods of over-temperature have little effect on insulation. Frequent over-temperature periods have the same effect as continuous over-temperature for that specific period of time. As can be seen from (Figure 3-18), there are four basic classes of insulation in the motor. The classes correspond to the hottest temperature found in the motor. They are rated as:

1. Class A — 221°F - 105°C
2. Class B — 266°F - 130°C
3. Class F — 311°F - 155°C
4. Class H — 356°F - 180°C

When we consider the maximum temperature of a motor, a reference temperature must be used. The reference temperature is usually 72°F. This means that the maximum temperature rise in A, B, F, and H is about 149, 194, 239, and 284°F respectively.

Class A insulating materials include cotton, silk and enamel coatings. Until recently, this was the most common wire used and the least expensive.

Class B insulation is made up of mica, asbestos, glass fiber and others with suitable bonding substances. Most motors are now wound with class B insulation.

Class F is mica, glass fiber, asbestos and other materials with compatible bonding agents. Although this class is not used widely, it fills the gap between class B and H.

Class H insulation consists of materials such as silicone elastomer, mica glass, asbestos and other material with compatible bonding agents. This is the most expensive and will withstand the highest temperatures.

POWER CALCULATION

The required power of a motor can be easily found if a constant load is being continuously drawn from

the machine. However, when the power requirements of a machine are not constant, the selection of motor size offers more the opportunity to control both the initial as well as the operating costs of the system.

When a motor is operated in a cyclic manner, the average heating must be found from motor losses during various parts of cycle. Also, the change in ventilation characteristics must be considered. The way of estimating the required size of a constant speed electric motor is by considering the armature heating. It may be assumed that heating can be determined from the variable horsepower losses and thusly depend on the square of the power output. As with all periodically varying currents, the average heating may be reflected by finding the RMS value of the power-time curve and choosing a motor of the least rating. The formula becomes:

$$\text{Power RMS} = \sqrt{\frac{\sum_{n=1}^{\infty} Pn^2 \times tn}{\sum_{n=1}^{\infty} tn}}$$

(3-4)

Where P is the power and t is the time that power is being drawn.

Considerations must be given to the maximum torque (power) limitations of a motor. Since RMS only considers heat dissipation characteristics of a motor, a final calculation as to peak torque must be made. For a standard Nema B motor, the maximum power during a peak will be limited to 150%. If peaks over 150% exist, a larger motor of Nema B type design or a special purpose high torque motor must be used.

EXAMPLE: (3-2)

A particular motor application has the following cycle: What size motor is needed?

1. 50 hp for 10 sec.
2. 150 hp for 5 sec.
3. 130 hp for 25 sec.
4. Idle for 20 sec. with motor running

SOLUTION:

Employing (3-4):

$$\text{Power RMS} = \sqrt{\frac{\sum_{n=1}^{\infty} Pn^2 \times tn}{\sum_{n=1}^{\infty} tn}}$$

$$\text{HP RMS} = \sqrt{\frac{(50)^2 x10+(150)^2 x5+(130)^2 x25+(0)^2 x20}{10 + 5 + 25 + 20}}$$

HP RMS = 96.6 HP

This means we select the next standard motor larger then or equal to 96.6. The next standard motor is a 100 HP. We must then check to see if the 100 HP motor will pull the peak horsepowers neded for the cycle. We multiply our standard motor by 1.5 which is a conservative estimation of its maximum torque. This value is then compared to the maximum horsepower required in the system.

Peak HP = Motor HP x 1.5

Peak HP = 100 x 1.5

Peak HP = 150 HP

This value is equal to the peak horsepower needed by the circuit, therefore the 100 HP motor will work.

CONCLUSION

In this chapter, we have covered the electric motor. Two formula's were introduced. They dealt with motor speed and RMS horsepower.

$$N\ (RPM) = \frac{120 \times \text{line frequency (Hz)}}{\#\ \text{of poles}}$$

$$\text{Power RMS} = \sqrt{\frac{\sum_{n=1}^{\infty} Pn^2 \times tn}{\sum_{n=1}^{\infty} tn}}$$

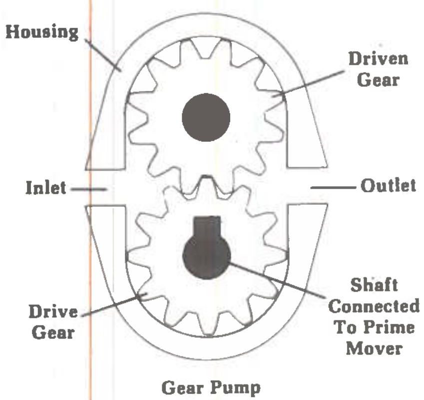
Many factors enter into deciding the type and size of an electric motor. Areas such as wire installation, NEMA class rating, enclosure and frame size must be considered.

Horsepower should be calculated using the RMS formula. Through its use, the motor size will be smaller and will be used more efficiently.

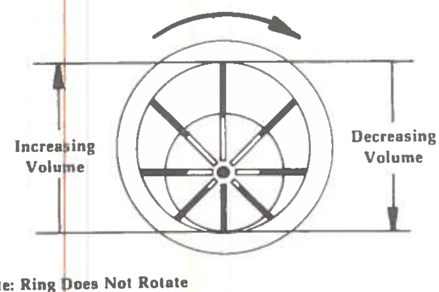
CHAPTER 4
SOURCES OF POWER
PUMPS AND ACCUMULATORS



(Figure 4-1)



(Figure 4-2)



(Figure 4-3)

The main source of fluid power for hydraulic systems is the hydraulic pump with its prime mover. The prime mover may vary from a fractional horsepower electric motor, waterfall, windmill, to a large internal combustion engine. Without the prime mover, the pump cannot provide hydraulic energy. Pumps used in fluid power applications are of the positive displacement variety which means that a nearly fixed volume of oil is displaced per revolution of the unit, regardless of changes in the pressure which the pump is developing. The difference between the actual output and the calculated output is denoted as volumetric efficiency.

$$\%\text{Volumetric Efficiency} = \frac{\text{Actual delivery per unit time}}{\text{Calculated delivery per unit time}} \times 100$$

The pump's mechanical efficiency depends on the frictional forces acting on moving parts.

$$\%\text{ Mechanical} = \frac{\text{Output Work done per Rev.}}{\text{Input work done per Rev.}} \times 100$$

The volumetric efficiency decreases with overall system pressure. The mechanical efficiency remains relatively constant throughout the pump range.

The overall efficiency is the product of the mechanical and volumetric efficiency.

$$\text{Overall efficiency} = \text{Mechanical efficiency} \times \text{Volumetric efficiency}$$

Let's take a look at the different types of pumps that exist and compare their performances.

GEAR PUMPS

Simple gear pumps are generally the least expensive for medium pressure applications. They are simple in construction consisting of a housing with inlet and outlet ports, and a pumping mechanism of two gears. It is the meshing and unmeshing of the gear teeth that cause the pump to work. The volumetric efficiency is good under optimum conditions, but tends to be poor at low speeds. It is classified as a fixed displacement pump because the output volume delivered, depends upon the pumps rotational velocity.

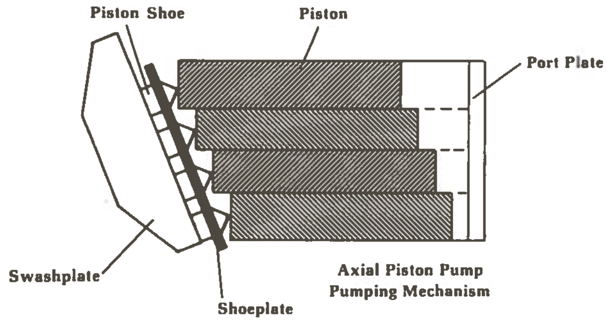
VANE PUMPS

Vane pumps generate a pumping action by causing vanes to track along a ring. As the space enclosed by the vanes, rotor, and housing enlarges, a vacuum is created and liquid is forced into the space by atmospheric pressure. As the space between the vanes decreases, fluid is forced out at the outlet. Volumetric efficiency tends to be better than gear

pumps. With this increase in performance, the cost must also increase.

PISTON PUMPS

Piston pumps generate a pumping action by causing pistons to reciprocate within a piston bore. For axial piston pumps, the amount of fluid displaced in a revolution can be controlled by the angle of the swashplate. Piston pumps are most suitable for high pressure applications, where sealing becomes a problem. This type of pump is of a more complex design with closer tolerances, hence the cost is higher than comparable pumps of equal displacement.

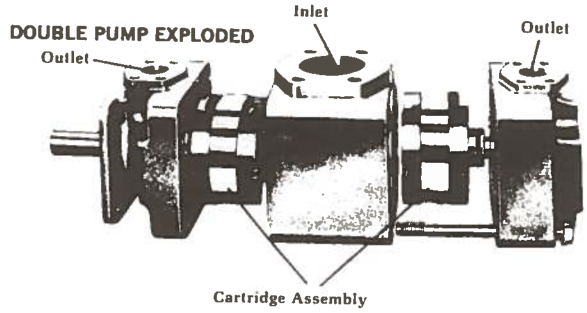


(Figure 4-4)

DOUBLE PUMPS

Double pumps usually exist in both gear and vane designs. A simple pump, as was described previously, consisted of one inlet and one outlet. A double pump consists of a housing with two separate pumping mechanisms.

There may be one or two inlets, and two separate outlets. In other words, a double pump is two single pumps in one housing. This pump can deliver two different flow rates, one from each outlet. Since both pumping units are connected internally, only one prime mover need be used.



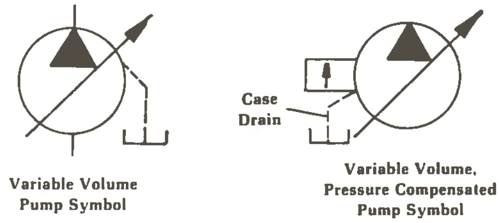
(Figure 4-5)

VARIABLE DISPLACEMENT PUMPS

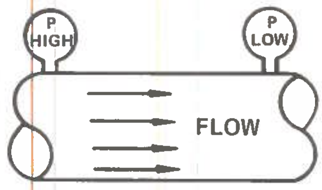
As was described before, a fixed displacement pump delivers a constant volume of fluid at a given pressure and rotational velocity. Fixed displacement type design is common to all three types of pumps. However, in some cases it is desirable that a pump's flow rate be variable. This can be readily accomplished in both vane and piston pumps. Although the input speed to the pump is constant, the output flow rate can be controlled to suit the needs of a circuit. Generally, a variable volume pump is pressure compensated.

A pump which is pressure compensated decreases fluid displacement to match the circuit demand at a pressure level. System pressure is limited by the setting of the compensator. Through the use of the type of pump, system horsepower can be conserved, because it causes less heat generation thus increasing system efficiency.

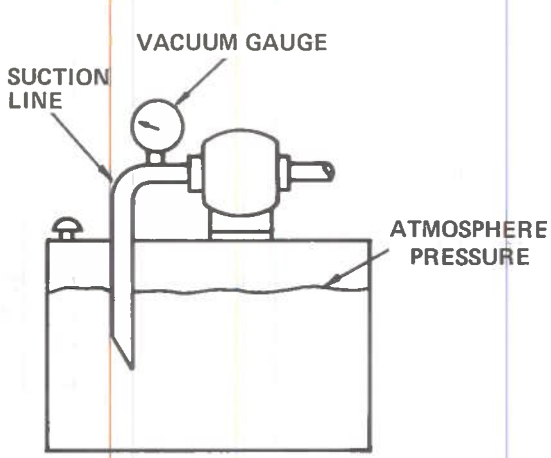
When we deal with pumps, they must be properly selected so that they can achieve the necessary flow during their cycle. They also must be capable of functioning properly within the maximum pressure of the system.



(Figure 4-6)



(Figure 4-7)



(Figure 4-8)

THE SUCTION SIDE OF THE PUMP

In order for a pump to deliver the necessary flow enough oil must enter the suction line of the pump. This is generally no problem when the pump is mounted below the fluid level and there is no restrictions to the flow of fluid. This is called flooded suction. This means that the pump needs to supply a minimum of energy to draw the fluid into the inlet. Gravity causes the fluid to enter the inlet. But what happens when a pump is mounted on top of the reservoir?

With the pump on top of the reservoir, energy must be exerted on the fluid in order to draw it into the pump. This energy will be in the form of a lower then atmospheric pressure (vacuum) induced at the inlet side of the pump. This is needed for the following reason:

In order to have flow, a pressure drop must exist. In (Figure 4-7) flow will exist from left to right, because the pressure at the gage on the right is lower than that on the left.

With the pump on top of the reservoir, the following representation can be made. (Figure 4-8)

As can be seen from the drawing, in order for flow to exist, the gage at the pump must be lower than 14.7 psia. The pump is quite capable of lowering this pressure, but it must not use an excessive amount of this energy drawing the fluid into the inlet.

You see the pump uses this lowering of pressure at the inlet to accomplish two phases.

Phase 1 — Supplying the liquid to the inlet.

NPSH — Accelerating the liquid and filling the rapidly moving rotating group (the pumping section)

NPSH (net positive suction head) is dependent on the type, the displacement and the RPM of the pump.

Phase (1) is dependent on the size, height, and configuration of the inlet line. The largest portion of the energy available on the suction side of the pump (limited by atmospheric pressure) is used in accelerating the liquid into the pump cavities. However, the action of supplying the pump with liquid takes place first. If too much is used in this phase, not enough energy is left to accelerate the liquid into the rotating group. This may cause the flow to the pump to diminish and pump failure will probably follow due to cavitation. To avoid destroying the pump, we must control the inlet conditions. First, one must consider the maximum energy available for the two phases.

The maximum energy a pump can deliver for phase (1) and NPSH is the atmospheric pressure available at the site where the pump is being used. This is because the pump can at best achieve an almost perfect vacuum at its inlet. We must now proportion this energy between phase (1) and NPSH.

From catalog data for a particular pump, we find the following statement:

> Inlet conditions: Not to exceed 7 inches. Hg. Vacuum on petroleum base fluids. Not to exceed 3 inches. Hg. vacuum on fire resistent fluids.

What the data means is that NPSH must be at least 29.92 inches Hg. (from this point on considered 30 in Hg. for standard atmospheric conditions) minus the rating (7 inches Hg.) for a petroleum base fluid. Or, in other words, for this particular pump — NPSH is:

$$\text{Atmospheric pressure at sea level} - \text{min. inlet pressure} = \text{NPSH}$$

(4-3)

$$\text{30 in Hg. sea level pressure} - \text{7 in. Hg.} = \text{23 in. Hg.}$$

To accelerate the fluid into the rotating group, for a fire resistant fluid, more energy is needed for the NPSH or:

$$\text{30 in. Hg. sea level pressure} - \text{3 in. Hg.} = \text{27 in. Hg.}$$

After we accelerate the fluid into the rotating group, any leftover energy can be allocated to phase (1). The energy left to accelerate the liquid is dependent on the site in which the pump is being used. If the pump is used at normal sea level conditions, (barometer reading is 29.92 in. Hg.), the maximum power for phase (1) is:

$$\text{atmospheric pressure at site} - \text{NPSH} = \text{maximum suction gage reading}$$

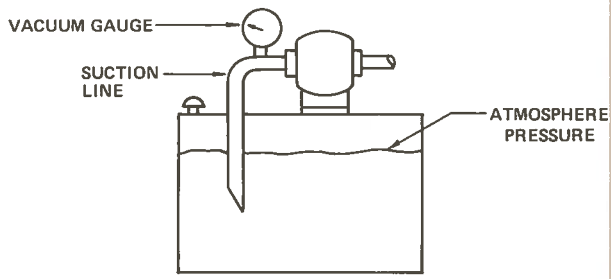
(4-4)

For a petroleum base fluid at sea level:

30 - 23 = 7 in. Hg. reading at a vacuum gage at the suction line.

If a gage at the suction line reads between 0 and 7 in. Hg. at sea level, the pump will function correctly. If the gage reads more than 7 in. Hg. something is incorrect. The nature of the problem must be corrected before the pump destroys itself.

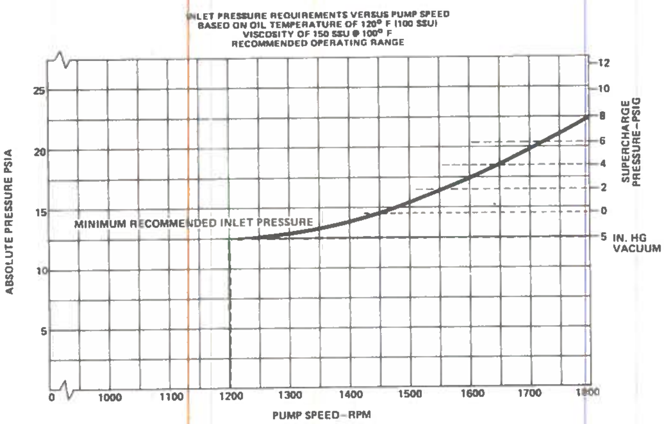
If the same pump is used in Denver, the atmospheric pressure may be 24.7 in. Hg. (as read from a barometer) the energy available for phase (1) can be calculated from formula (4-4) We find:



(Figure 4-9)

Altitude Above Sea Level in Feet	Barometer Reading — Inches of Mercury	Atmospheric Pressure — Pounds Per Square Inch
0	29.92	14.7
1000	28.8	14.2
2000	27.7	13.6
3000	26.7	13.1
4000	25.7	12.6
5000	24.7	12.1
6000	23.8	11.7
7000	22.9	11.2
8000	22.1	10.8
9000	21.2	10.4
10000	20.4	10.0

(Figure 4-10)



(Figure 4-11)

Atmospheric Pressure at site - NPSH = maximum suction gage reading

24.7 in. Hg. - 23 in. Hg. = 1.70 in. Hg.

At Denver, if the suction gage reads more than 1.70 in. Hg. the pump may be cavitating. If a fire resistant fluid is used in our Denver example, we find the energy for phase (1) equal to:

24.7 - 27 in. Hg. = - 2.3 in. Hg.

The answer obtained is negative. When this happens, we interpret it as meaning that the inlet of the pump must be pressurized at least 2.3 in. Hg. in order for this pump to function. It can be in the form of a flooded pump inlet, a pressurized reservoir or a suitable supercharge pump. (A supercharge pump is one which delivers fluid at a low pressure to the main pump in the system.) In either case, a vacuum would probably not be read at the suction side of the pump. An above atmospheric pressure would be read at the suction side of the pump.

Many times inlet conditions are given in terms of RPM's. If this is the case, just retrieve the corresponding maximum inlet condition from the curve and perform the preceding procedure:

EXAMPLE 1:

The pump X, operating at 1200 RPM, shows a vacuum at the suction line of 3 inches of mercury. The site at which the pump is running is 1000 ft. above sea level. Is the vacuum within the limitations of the pump?

SOLUTION:

Two items must be obtained:

Item 1 — The atmospheric (barometer reading) pressure at 1000 ft.

Item 2 — The inlet characteristics of the pump.

Item 1 can be obtained from the following chart:

(Figure 4-10)

At 1000 ft. above sea level, the atmospheric pressure is:

Item 1 is 28.8 in. Hg.

Item 2 can be obtained from the manufacturer's catalog. For pump x the following was obtained (Figure 4-11).

At 1200 RPM the minimum inlet pressure is 5 in. hg.

Employing formula (4-3) we can find the energy necessary for phase NPSH.

Atmospheric pressure at sea level - minimum inlet pressure = NPSH

30.0 - 5.0 = 25.0

We need 25.0 in. Hg. to accelerate the fluid into the rotating group. Employing formula (4-4) we can find the maximum suction gage reading at the gage.

Atmospheric pressure gage at site - NPSH = maximum suction reading

Since the gage reads 3 in. Hg. (this value being lower than 3.8 in. Hg.) the pump is working fine.

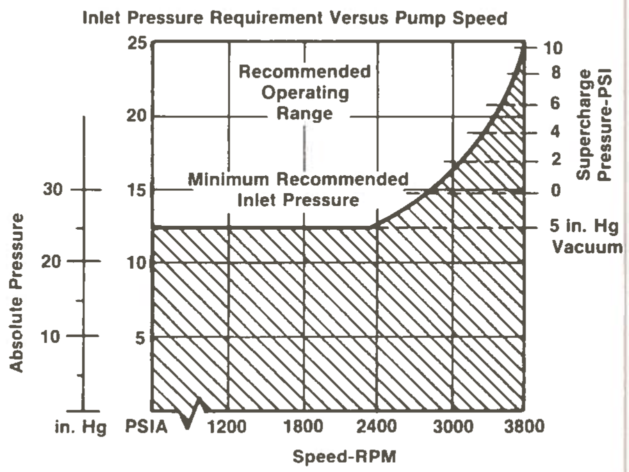
WHY DIFFERENT SPEEDS ON MAXIMUM ALLOWABLE VACUUM

A maximum allowable vacuum specification given by a manufacturer is affected by the speed at which it is turned by its prime mover.

A pump having an allowable vacuum of 5 in. Hg. @ 1200 RPM indicates that the manufacturer desires a minimum pressure of 25 in. Hg. absolute pressure at pump inlet to accelerate liquid into the pump. With the same pump operating at 1800 RPM, the fluid must move faster since less time is available for filling. This action requires more pressure: consequently, maximum allowable vacuum is decreased. On the other hand, a pump rated at a maximum allowable vacuum of 5 in. Hg. @ 1800 RPM, would have an increased allowable vacuum at 1200 RPM.

As pump speed is increased, maximum allowable vacuum is decreased. To illustrate this point, catalogs of some pump manufacturers describe required inlet pressure vs. speed as a curve; such a curve is illustrated. The pump may be operated within the clear area.

It will be noted that vacuum and absolute pressure are on the same axis.

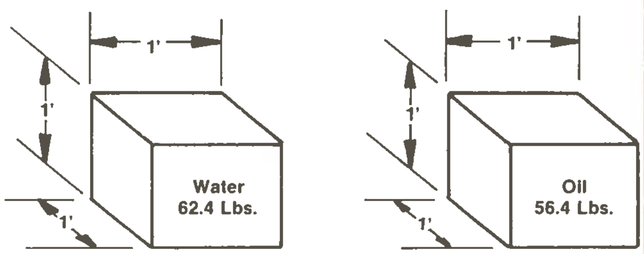


(Figure 4-12)

EFFECTS OF DIFFERENT FLUIDS ON MAXIMUM ALLOWABLE VACUUM

Specific gravity compares the weight of one liquid with another. More specifically, it is the ratio of weight between a volume of water and an equal volume of another liquid.

A cubic foot of water weighs 62.4 lbs. @ 60° F. A cubic foot of common petroleum base hydraulic fluid weighs 56.4 lbs. @ 60° F. Dividing the weight of the hydraulic fluid by the water weight, we find that the weight of the oil is 87% of the water. The ratio of weight is 1 (water) to .87 (petroleum oil). The



(Figure 4-13)

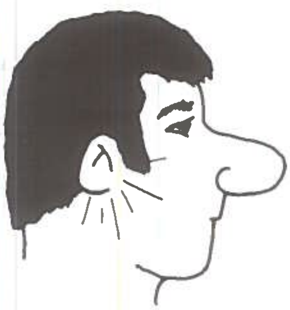
Typical Vapor Pressures of Water Glycol Fluids

Temp. °F.	Vapor Pressure (in. Hg)
100	1.3
110	1.9
120	2.5
130	3.3
140	4.1
150	5.5

Typical Vapor Pressures of Water-In-Oil Emulsion Fluids

Temp. °F.	Vapor Pressure (in. Hg)
100	1.9
110	2.6
120	3.5
130	4.5
140	5.9
150	7.6

(Figure 4-14)



(Figure 4-15)

specific gravity (SG) of petroleum oil is then indicated as .87.

A pump's maximum allowable vacuum specification is based on using petroleum base fluid which has a specific gravity of approximately .87. With a phosphate ester fire resistant fluid, specific gravity increases over 30% to approximately 1.15. Water base fluids have specific gravities ranging from .93 for an invert emulsion to 1.08 for water glycols. (See fluids chart in Design Engineer's Handbook.) With these heavier fluids, more pressure is required at pump inlet for fluid acceleration; consequently; maximum allowable vacuum may decrease appreciably.

HIGH VAPOR PRESSURE AFFECTS MAXIMUM ALLOWABLE VACUUM

Petroleum base and phosphate ester fire resistant fluids have very low vapor pressures at the operating temperature of a common hydraulic system. However, this is not the case with a water base fluid.

Since they contain a large percentage of water, invert emulsions and water glycol fluids can have vapor pressures of several inches of Hg where petroleum and synthetic fluids would have vapor pressures of a fraction of an inch of Hg. For this reason, they have more of a tendency to vaporize and cavitate.
To avoid cavitation with a water base fluid, a pump manufacturer ensures that sufficient pressure is present at pump inlet to accelerate the liquid into the pump without ever going below the fluid's vapor pressure. This is accomplished by reducing maximum allowable vacuum.

CHECK FOR PUMP CAVITATION

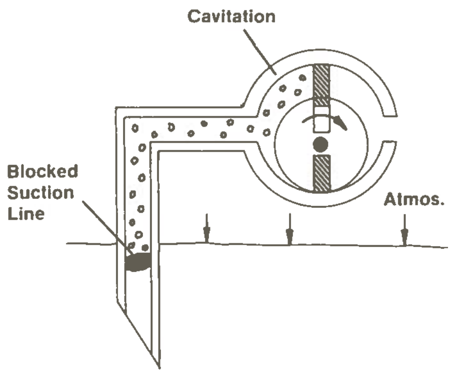
Maintenance men have the first opportunity to discover a cavitating pump or a pump which is sucking air. Being acquainted with their machinery, they receive the initial indication that something is wrong.

The most pronounced indication of a pump either cavitating or sucking in air are high-pitched sounds, but they are of a slightly different character. A cavitating pump will have a steady, high-pitched sound which is probably due to collapsing bubbles of approximately the same size. On the other hand, a pump sucking air has somewhat of an erratic sound. If small amounts of air are finding their way to the pump, the noise may sound like a rattle or a bad bearing. Large amounts of air will result in a very erratic banging and popping noise.

A more certain way to distinguish cavitation from air

entrainment is to determine the absolute pressure at pump inlet by taking a vacuum gage reading and subtracting it from the barometric pressure. If this action indicates that insufficient pressure is available at pump inlet, cavitation may be occurring.

A new system with a cavitating pump is probably the result of poor suction line design or incorrect fluid viscosity. Changing to the appropriate viscosity or increasing the suction line pipe size reducing pressure differential will help the situation. A properly designed older system whose pump is cavitating may be the result of a plugged suction line due to a rag, newspaper, or animal. It may also be caused by a dirty filter without a bypass or an insufficient bypass.



(Figure 4-16)

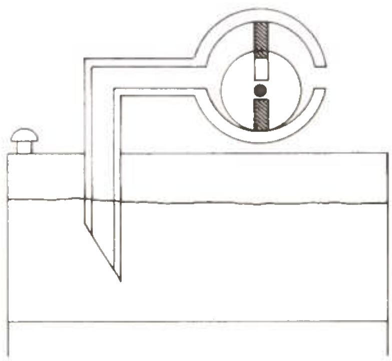
PUMP PRIMING

With respect to a pump, "primed" indicates that a pumping mechanism is filled with liquid. A pump which is not primed is filled with air or "air bound." Before pumping can occur, this air must be purged from suction line and pump cavities. If this is not done and the pump is allowed to operate in an unprimed condition for even a few minutes, it may be permanently damaged due to lack of lubrication.

At start up, a pump whose outlet is directly connected to tank through a directional valve, generally has an easy time of pushing trapped air to tank. A pump which must push its air over a relief valve has an impossible task. The reason being an ordinary industrial hydraulic pump is a poor air compressor.

To release trapped air from an uprimed pump, a fitting at pump outlet can be loosened. The pump is then jogged until oil squirts out the fitting indicating the pump is primed. The fitting is then tightened. Trapped air may also be released by venting a relief valve.

Pump priming is frequently required on new systems at start up and on systems with suction side leaks.



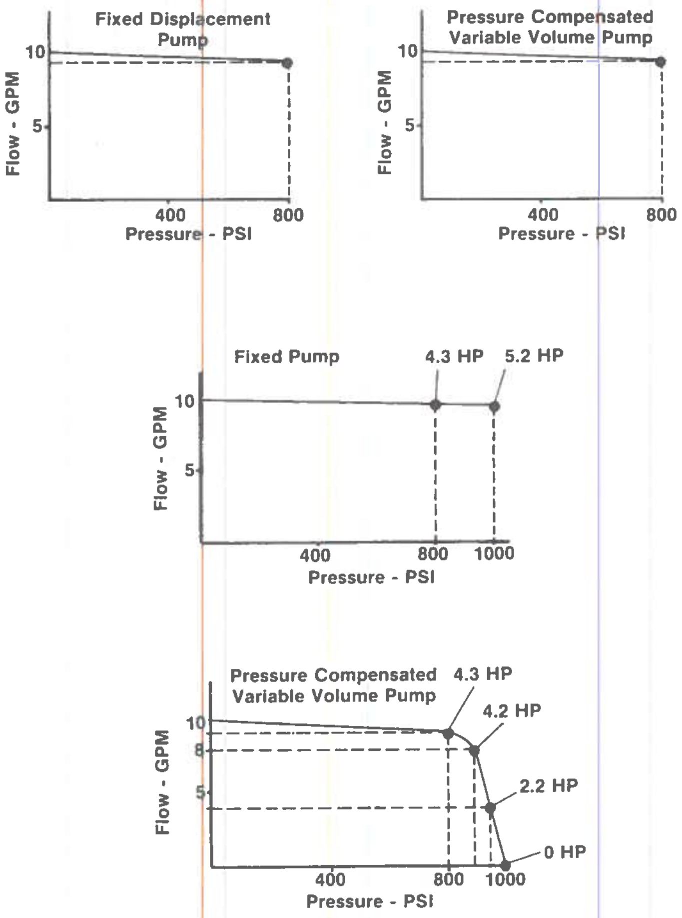
(Figure 4-17)

THE PRESSURE SIDE OF THE PUMP

Since the inlet conditions of the pump are satisfied, we are ready to discuss the pressure side of the pump. Here we will touch on the generation of hydraulic horsepower through the implementation of fixed displacement pumps, as well as pressure compensated variable volume pump.

COMPARING FIXED AND PRESSURE COMPENSATED VARIABLE VOLUME OPERATION

Illustrated on the page are two curves. One curve shows the discharge flow from a 10 GPM fixed displacement pump as it operates between zero and 800 PSI. The other curve shows the discharge flow from a 10 GPM variable volume pump as it operates between the same pressures. Both curves are the same and they point out that the discharge flow rate from each unit is at 800 PSI is approximately 9.25 GPM; this is 4.3 HP.



(Figure 4-18)

Assume that the fixed displacement pump's maximum pressure is limited by a relief valve setting of 1000 PSI. Referring to the fixed displacement curve between zero and 1000 PSI, we see that at 1000 PSI pump is developing 9 GPM which is 5.2 hydraulic horsepower. This is the maximum power that the unit is allowed to develop; and as it passes through a relief valve, it is transformed into heat.

Assume now that the pressure compensated variable volume pump has a compensator setting of 1000 PSI. Referring to the pressure compensated pump curve between zero and 1000 PSI, we find that at 800 PSI and 9.25 GPM (4.3 HP), discharge flow begins to decrease. Consequently, generated power decreases as system pressure approaches 1000 PSI. At a pressure of 900 PSI, discharge flow is 8 GPM (4.2 HP). At 950 PSI, discharge flow is 4 GPM (2.2 HP). And at 1000 PSI, discharge flow rate is zero GPM (0 HP). Horsepower considered here is power delivered to the system.

Pressure compensated vane pumps are characteristically fully compensated within 200 PSI. Piston pumps compensate fully within 100 PSI or less. Since a vane pump has a wider compensating band, the curves of pressure compensated, variable volume vane pump will be shown. This will more easily illustrate a pump's operating characteristic during compensation.

With a pressure compensated pump coupled to an electric motor, generated hydraulic power is not only controlled by limiting pressure, but it is reduced by regulating discharge flow. This means less heat is generated in a system since generated power is not wasted over a relief valve.

INPUT POWER FOR A FIXED DISPLACEMENT PUMP

The power delivered by the pump is proportional to the flow rate times the pressure at any given time in the cycle. The equation was given in chapter (2) is:

HP system = GPM x PSI x .000583
(4-5a)

OR

P (KW) = (LPM x bar) / 600
(4-5b)

To calculate the power that need be delivered by the motor we must divide equation (4-5) by the efficiency.

$$\text{Input P} = \text{P motor} = \frac{\text{P system} \times 100}{\text{Overall Efficiency (\%)}}$$
(4-6)

NOTE: Many times the input P is given and need not be calculated.

The efficiency of the pump is a factor dependent on the flow rate and pressure and must be obtained from manufacturers data.

The heat generated by the pump due to efficiency is:

Heat Generation Rate (P) = (P motor - P system)
(4-7)

Heat Generation Rate (BTU/hr) =
(HP motor - HP system) x 2547
(4-8a)

OR

Heat Generation Rate (J) =
(P motor - P system) /.4147
(4-8b)

$$\text{Average Heat} = \frac{\sum_{n=1}^{\infty} \text{Pn} \times \text{tn}}{\sum_{n=1}^{\infty} \text{tn}}$$
(4-9)

Many times important data about pumps is given in terms of a chart. If this is the case, this data must be reconstructed and put into a graphical form to be of good use.

The two most important parameters needed when dealing with pumps is the output volume and overall efficiency. Both of these must be plotted verses pressure.

The data usually given is output flow and input horsepower at two or three pressures. First, plot flow vs. pressure on an appropriate graph directly from given data. Next we will plot efficiency vs. pressure. Equation (4-6) may be rearranged so that overall efficiency can be calculated. To show the ease of using this technique, let us work out an example.

EXAMPLE: (4-2)

A pump has the following cycle:

1. Advance — full flow @ (100 psi) (6.9 bar) in 4 sec.
2. Feed — full flow @ (1800 psi) (124 bar) in 8 sec.
3. Return — full flow @ (100 psi) (69 bar) in 3 sec.
4. Idle — full flow @ (50 psi) (3.45 bar) in 10 sec.

   a. Calculate the heat generation rate
   b. Calculate the motor size

From a pump catalog, the following data was retrieved:

The first operation we must follow is to find the flow rate of the pump at various pressures. This is a simple matter of reading it directly from the chart. They are:

Pressure (bar)	6.9 bar	68.9 bar	172 bar
Pressure (psi)	100 PSI	1000 PSI	2500 PSI
Flowrate GPM/LPM	17.1 64.7	16.4 62	15.5 58.7
Power HP KW	1.42 1.06	11.58 8.48	27.6 20.58

The overall efficiency can be obtained by using equation (4-6). Rearranging it becomes:

$$\text{Overall efficiency} = \frac{\text{P system}}{\text{Input P}} \times 100$$

(4-6)

We may substitute equation (4-5a) for the system horsepower in equation (4-6) and it will become:

$$\text{Overall efficiency} = \frac{\text{GPM} \times \text{PSI} \times .000583}{\text{Input HP}} \times 100$$

Overall efficiency = (LPM x bar) / (600xP(KW)) x 100

We will use the above formula to find the overall efficiency at the three given pressures.

Overall efficiency @ 100 psi =

$$\frac{(17.1 \times 100 \times .000583)}{1.42} \times 100 = 70\%$$

Overall efficiency @ 6.9 bar =

(64.7 x 6.9) / (600 x 1.06) x 100 = 70%

Overall efficiency @ 1000 psi =

$$\frac{(16.4 \times 1000 \times .000583)}{11.38} \times 100 = 84\%$$

Overall efficiency @ 68.9 bar =

(62 x 68.9) / (600 x 8.48) x 100 = 84%

Overall efficiency @ 2500 psi =

$$\frac{(15.5 \times 2500 \times .000583)}{27.6} \times 100 = 82\%$$

Overall efficiency @ 172 bar =

(58.7 x 172) / (600 x 20.58) x 100 = 82%

The three flowrates and efficiencies are now ready to be plotted on a graph. Points will be connected with straight lines. Flow rate will be a maximum at zero psi, while efficiency must be zero at the same point. A graph as shown will result. See next page.

a. To calculate the heat generated, we must first calculate the system HP needs.

ADVANCE

Since 6.9 bar (100 psi) information is given, we may use the chart directly.

Using (4-5)

HP system = (LPM x bar) / 600
(64.7 LPM x 6.0 bar) / 600

HP system = .742 KW (.996 HP) for 4 seconds.

Input HP from chart is (1.42 HP) 1.06 KW at the above conditions.

Therefore, heat generated is:

Heat generated = P Motor - P system

Heat generated = 1.06 - 0.742 = 0.318 KW for 4 sec.

The efficiency of the pump can be obtained by reading the graph. It is found to be 70%.

FEED

Here we must use the graph. Entering at (1800 psi) 124 bar, we find the flowrate and efficiency is (15.8 GPM) 60 LPM and 83% respectively.

Using (4-5)
HP = (LPM x bar) / 600

= (60 LPM x 124) / 600
= 12.4 KW (16.58 HP) for 8 seconds

Using (4-6)

$$\text{Input P} = \text{P motor} = \frac{\text{P system}}{\text{Efficiency}} = \frac{12.4}{83} \times 100$$

P motor = (19.97 HP) 14.4 KW for 8 sec.

Heat generated is P motor - P system

Heat generated = 14.4 KW - 12.4

Heat generated = (3.39 HP) 2 KW for 8 sec.

RETURN

Return power is the same as advance power, except for the time. The values are just-copies from advance section.

HP system = (0.996 HP) 0.742 KW for 3 sec.

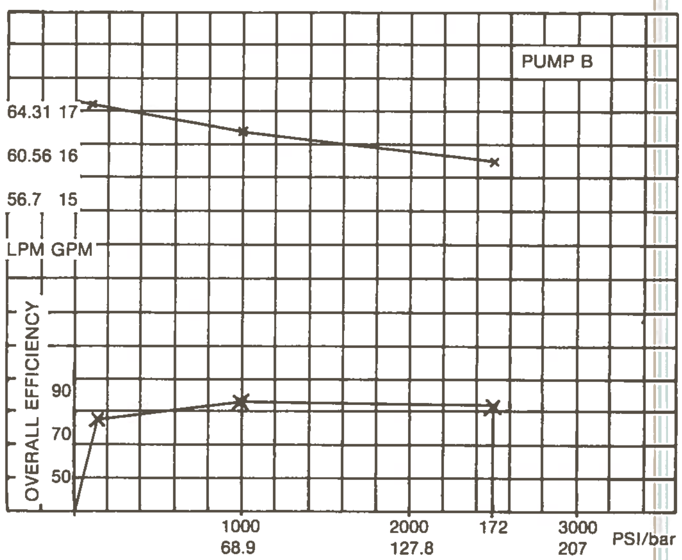
Input HP = (1.42 HP) 1.06 KW for 3 seconds.

Heat generated = (.426 HP) .318 KW for 3 seconds.

IDLE

Information during idle must also be obtained from the graph. The flowrate and efficiency is 65.1 LPM and 40% respectively.

	P System KW/HP	P Pump KW/HP	Heat Generation Rate KW/HP	Duration	Pump Efficiency
Advance	.742/.996	1.06/1.42	0.318/.426	4	.70
Feed	12.4/16.58	14.4/19.97	2/3.39	8	.83
Return	.742/.996	1.06/1.42	0.318/.426	3	.70
Idle	.374/.5	.935/1.26	.561/0.76	10	.4



Idle horsepower is obtained from (4-5)

HP system =(LPM x bar) / 600

HP system = 0.374 KW (.540)

Using (4-6) the input power is:

$$P \text{ Input} = \frac{P \text{ system}}{\text{Efficiency}} = \frac{.374}{40} \times 100 = .935 \text{ KW}$$

The average heat generated is:

$$\text{Heat Generated Average} = \frac{\sum_{n=1}^{\infty} Pn \times tn}{\sum_{n=1}^{\infty} tn}$$

$$= \frac{0.318 \times 4 \text{ sec.} + 2 \times 8 \text{ sec.} + 0.318 \times 3 \text{ sec.} + .56 \times 10 \text{ sec.}}{25 \text{ seconds}}$$

= .95 KW generated/cycle

The motor size can be obtained using formula (3-2)

$$P \text{ RMS} = \sqrt{\frac{\sum_{n=1}^{\infty} Pn^2 \times tn}{\sum_{n=1}^{\infty} tn}}$$

$$= \sqrt{\frac{(1.06)^2 \times 4 + (14.4)^2 \times 8 + (1.06)^2 \times 3 + (0.935)^2 \times 10}{25}}$$

= 8.44 KW or 11.32 HP

A 15 HP motor should be adequate in this application. No trouble should be encountered if pump data is given in graphical terms. Also, when a double pump situation is found, both flow rates and pressures being delivered by the pump must be used to calculate the horsepower delivered by the motor.

INPUT POWER FOR A PRESSURE COMPENSATED VARIABLE VOLUME PUMP

With a pressure compensated pump, the procedure discussed previously will be employed for all points except during a period of partial or full compensation.

During full compensation, the power that must be delivered to a particular pump is dependent on pressure and its rotational speed. This input power is required to overcome the mechanical inefficiency and displace leakage flow at the compensated pressure. The value for power can be read directly from catalog data. This will be found at a curve reading — input horsepower - pump compensated.

To obtain the input power during partial compensation, consider the following. Power is delivered to the pump and proportioned to two different areas.

First, it is proportioned to the flow displaced to the system. This is calculated using equation (4-5).

P system = System flow x Pressure x Constant

Secondly, power is proportioned to make up leakage and overcome mechanical inefficiency. This value is obtained by reading directly from the pump compensated curve. Adding the two, we can obtain the necessary power that must be delivered during partial compensation.

P Partial Compensation = Flow x Pressure x constant + input P at full compensation

= GPM x PSI x .000583 + Input HP at full comp. (4-10a)

= LPM x bar ÷ 600 + Input power (KW) @ full comp. (4-10b)

EXAMPLE: (4-3)

A particular pressure compensated pump is used as follows: (Figure 4-19).

Find the motor horsepower.

	GPM	PSI	TIME (SEC.)
PART 1	10	900	2
PART 2	9.6	1500	7
PART 3	4	?	3
PART 4	0	2000	2

(Figure 4-19)

SOLUTION: Input power to the pump must be calculated.

Catalog data is given in (Figure 4-20)

From the curves, the input horsepower for part 1 and 2 are read directly.

Part 1 HP input full flow = 6.3 HP
Part 2 HP input full flow = 10.5 HP

For Part 3, we first must find the pressure at which 4 GPM will be delivered. Finding the 2000 PSI compensation curve, we follow it until it intersects the 4 GPM line. The value obtained is approximately 2000 psi.

Using equation (4-10a)

HP Partial compensation = HP system + Input HP at full compensation

HP system = GPM x PSI x .000583

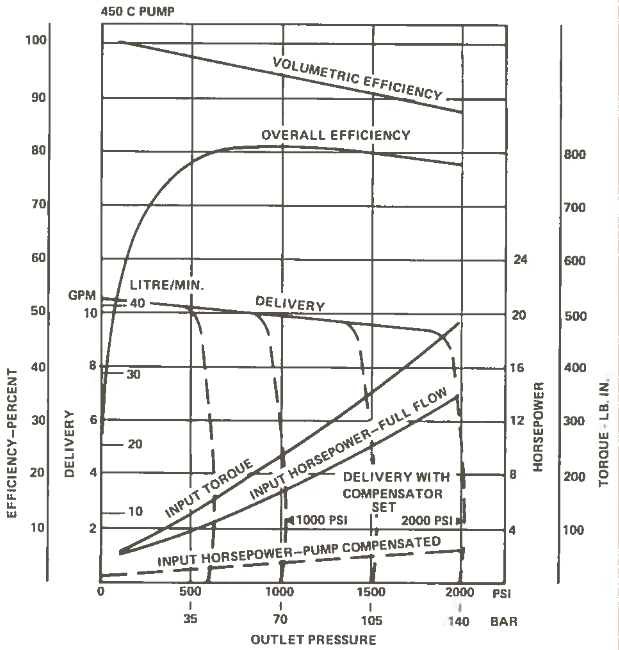
= 4 x 2000 x .000583

HP System = 4.664 HP

Input HP at full compensation is read directly from the curve. It is found to be 2.9 HP. Total HP for part 3 is:

HP = 2.9 + 4.664 = 7.564 HP

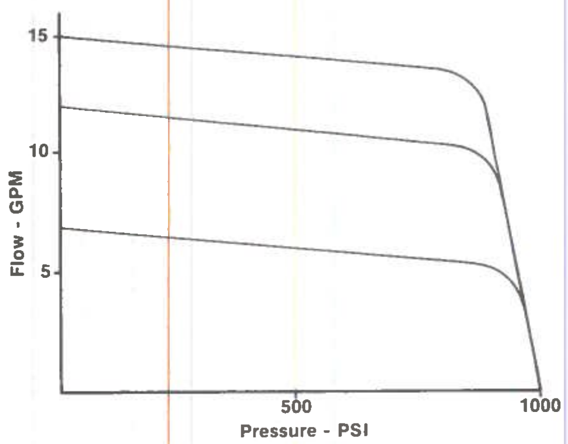
Part 4 is read directly from the chart. It is found



(Figure 4-20)

	HP INPUT	DURATION
PART 1	6.3	2
PART 2	10.5	7
PART 3	7.564	3
PART 4	2.9	2

(Figure 4-21)



(Figure 4-22)

where the line-input horsepower pump compensated intersects the compensated pressure line of 2000. The value is:

HP input = 2.9 HP

The data obtained in chart form is shown in (Figure 4-21)

The motor horsepower is:

$$HP\ RMS = \sqrt{\frac{\sum_{n=1}^{\infty} Pn^2 \times tn}{\sum_{n=1}^{\infty} tn}} =$$

$$\sqrt{\frac{(6.3)^2 \times 2 + (10.5)^2 \times 7 + (7.564)^2 \times 3 + (2.9)^2 \times 2}{14}}$$

HP RMS = 8.57

Motor size would be a 10 HP.

Pressure compensated vane pumps are acharacteristically fully compensated within 200 psi. Piston pumps compensate fully within 100 PSI or less. Since a vane pump has a wider compensating tract, the curves of pressure compensated variable volume vane pumps will be shown. This will more easily illustrate a pump's operating characteristics during compensation.

With a pressure compensated pump coupled to an electric motor, generated hydraulic power is not only controlled by limiting pressure, but it is reduced by regulating discharge flow. This means less heat is generated in a system since generated power is not wasted over a relief valve.

We have seen that when a pressure compensated pump is fully compensated, the generated hydraulic power to the system is zero. However, the electric motor coupled to the pump must still develop power because of internal leakage and mechanical inefficiencies of the pump.

ADJUSTING THE MAXIMUM VOLUME OF A VARIABLE VOLUME PUMP

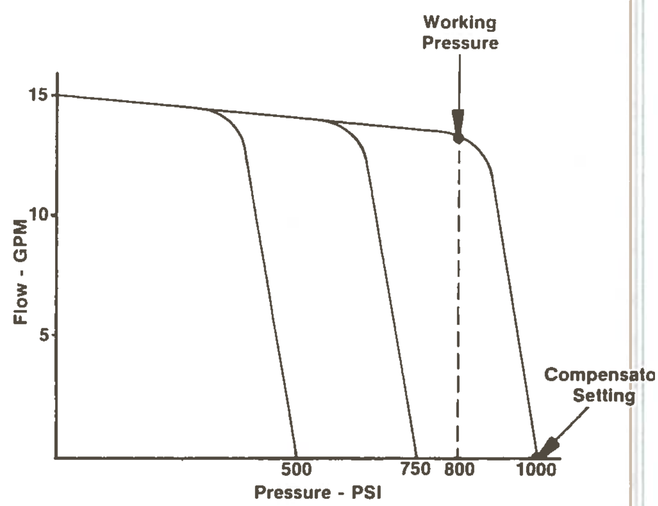
Setting a maximum volume adjustment determines at what flow point pump operating will begin. Illustrated is the performance curve for a 15 GPM pressure compensated, variable volume pump. With the volume adjustment backed out and the compensator set for 1000 PSI, the curve starts at a flow rate of 15 GPM. With the adjustment screwed in slightly, the pump curve starts at 12 GPM. If adjusted in more, pump operation begins at 7 GPM.

Variable volume pumps are manufactured in various size ranges like 5 GPM, 10 GPM, 15 GPM, 20 GPM, 30 GPM, 50 GPM, etc. If a system requires a 12 GPM, it would be normal to use a 15 GPM unit adjusted to 12 GPM.

With a variable volume pump, discharge flow can be adjusted from its maximum volume to anything below. In this way a pump is not required to generate excess flow rate and power unnecessarily. With a variable volume pump, power generation can be more evenly matched with actuator power output.

ADJUSTING THE PRESSURE COMPENSATOR OF A VARIABLE VOLUME PUMP

Properly adjusting the compensator of a pressure compensated, variable volume pump involves a knowledge of the pump working pressure and the pump performance curve.



(Figure 4-23)

Working pressure for a pump indicates the amount of pressure which it must develop to overcome resistances of load and liquid. The setting of a pressure compensator should be above this pressure while work is being performed.

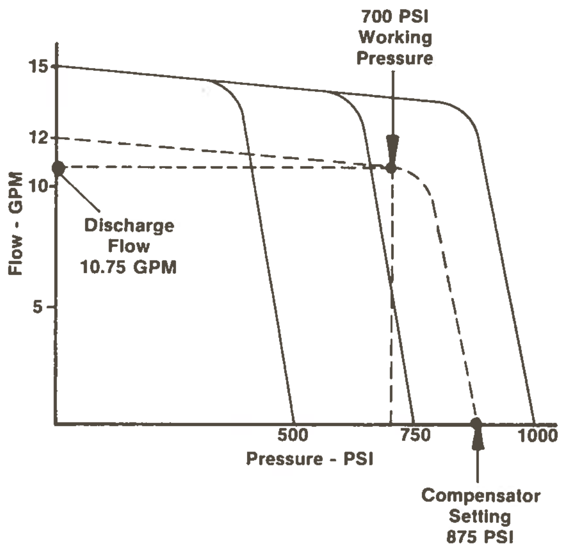
From the illustrated performance curve of a pressure compensated variable volume vane pump, it can be seen that discharge flow rate remains relatively constant until 200 PSI before the compensation point. This means that the compensator setting must be at least 200 PSI above pump working pressure — when operating at maximum volume.

For each type of pressure compensated, variable volume pumps, a performance curve is published. This curve illustrates pump performance at maximum volume at various compensator settings. If the volume adjustment is set at something less than maximum, a line is drawn on the graph which starts at the volume adjustment setting and parallels the horizontal portion of the performance curve.

With working pressure known and the pump volume curve drawn, compensator setting can then be determined.

EXAMPLE: (4-4)

Illustrated is a performance curve for a 15 GPM pressure compensated variable volume vane pump. Assume that the pump is operating in a circuit with one cylinder and that it is adjusted for a maximum flow rate of 12 GPM. On the curve a line is drawn which starts at the maximum adjusted volume of 12 GPM and parallels the flat portion of the curve starting at 15 GPM.

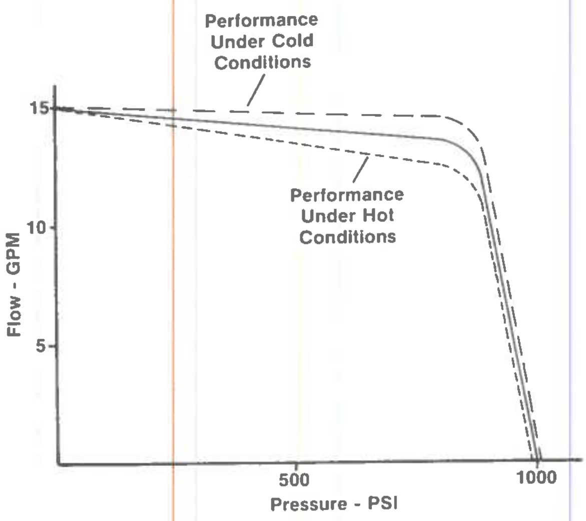


(Figure 4-24)

With a compensator adjusted so that it does not interfere with discharge flow during work, and with system heated to operating temperature, the cylinder is cycled through its work stroke. A gage at pump outlet indicates 700 PSI at its highest point, this is

pump working pressure. On the graph, a line is drawn from the 700 PSI point on the pressure line upward to the horizontal part of the curve. It intersects the line where flow is 10.75 GPM; this is 4.4 HP. Starting at this point, a line is drawn which parallels the compensator curve to the right. This line intersects the pressure line at 875 PSI. This should be the minimum compensator setting.

With pump deadheading, the compensator is then adjusted so that the gage reads at least 875 PSI.



(Figure 4-25)

VARIABLE VOLUME PUMP SETTINGS AFFECTED BY TEMPERATURE

As described, pressure compensated, variable volume pumps have adjustments for maximum volume and maximum pressure. Both of these settings are affected by temperature.

Assume that a variable volume, pressure compensated, pump is adjusted for a maximum volume of 15 GPM. While system is running at its operating temperature, a portion of the 15 GPM leaks back to inlet. This leakage is common and is due to the pump's volumetric efficiency.

If system temperature increased, viscosity would decrease causing more fluid to leak back to inlet. Pump discharge flow would decrease. If system temperature decreased, viscosity would increase causing less fluid to leak back to inlet, discharge flow would increase. The setting of the maximum volume adjustment is affected by temperature. If discharge flow is critical to machine performance, a variable volume pump should be adjusted and operated at system operating temperature.

A pressure compensator of a variable volume pump is also affected by temperature.

Assuming no leakage is present in the pressure side of the system, no flow discharges from pump outlet as a pump is fully compensated. However, because of internal pump leakage, pump displacement stabilizes to make up for leakage. This might mean that as a pump compensates at 1000 PSI, no flow discharges from pump outlet, but its pumping mechanism is displaced so that leakage is made up.

When temperature increases so does internal leakage through the pump. This results in more displacement from the pumping mechanism during compensation which means the force biasing the swashplate or cam ring is less than under reduced leakage conditions. Therefore, the no-flow compensation pressure will be lower under increased temperature conditions.

The opposite occurs when the fluid cools and viscosity increases. This reduces leakage causing the pumping mechanism to be displaced even less under full compensation. Pressure biasing a swashplate or cam ring will be more to achieve an equilibrium condition. Consequently, the no-flow compensation pressure will be higher under decreased temperature conditions.

If full compensator pressure is critical to machine performance, a pressure compensated pump should be adjusted and operated at system operating temperature.

Now let's see how a system is affected by wear of a pressure compensated variable volume pump.

EFFECTS ON WEAR ON PRESSURE COMPENSATED VARIABLE VOLUME PUMP PERFORMANCE

After a period of time, the rotation elements of a pump wear. This affects the performance of a pressure compensated, variable volume pump from the aspect of discharge flow and no-flow compensating pressure.

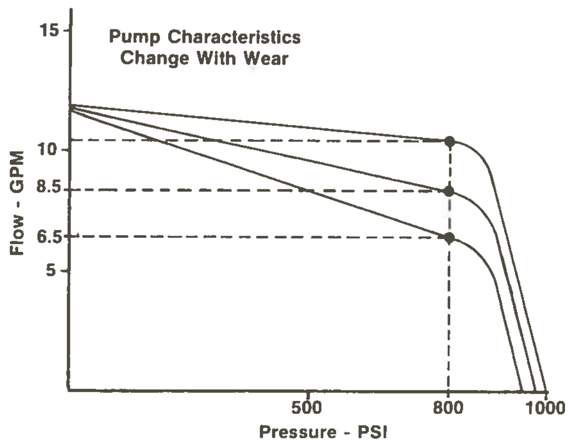
Illustrated is a performance curve for a 15 GPM pressure compensated pump. The pump has been adjusted for a maximum flow rate of 12 GPM and a compensator setting of 1000 PSI. It can be seen from the curve that at 800 PSI flow rate is 10.5 GPM and at 1000 PSI discharge flow is zero.

As the pump wears, leakage increases at every point along the curve. At 800 PSI, flow rate is no longer 10.5 GPM, but 8.5 GPM. And, the pump is fully compensated at 975 PSI instead of 1000 PSI. The characteristics of this pump have been changed; the curve has shifted.

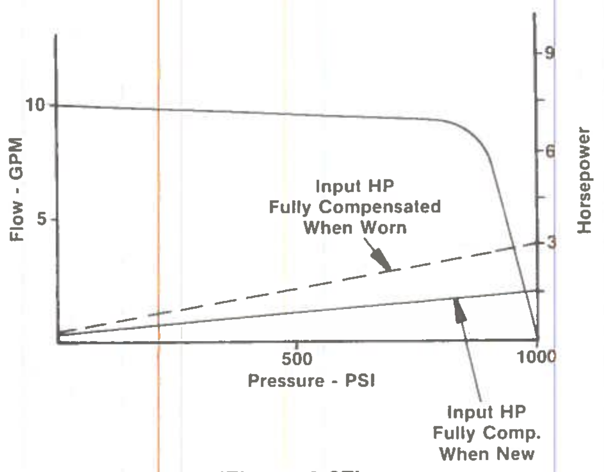
If pump wear increases, flow at 800 PSI, might be 6.5 GPM. Now the pump would be fully compensated at 950 PSI.

With a fixed displacement pump, nothing can be easily done to make up for increased wear. With a variable volume pump, the pump volume adjustment can be backed out of where is any adjustment left.

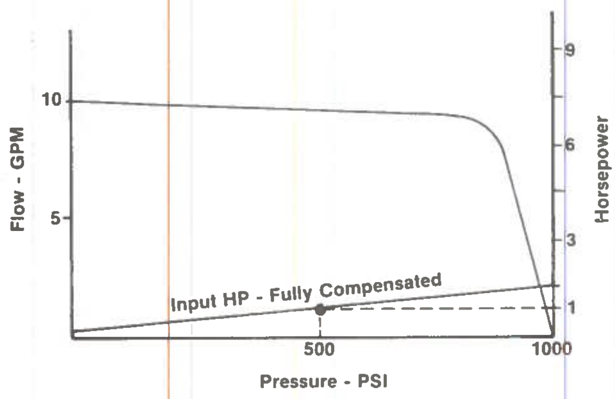
With the pump adjusted for 10.5 GPM at 800 PSI, but because of wear 8.5 GPM discharges from the pump at 800 PSI, the maximum volume adjustment can be backed out to make up for the increased leakage. Now the pump delivers 10.5 GPM at 800 PSI (4.9 HP) just as under original conditions. However, since internal leakage is so great, the electric motor works harder and the pump runs hotter due to the wasted power caused by the leakage.



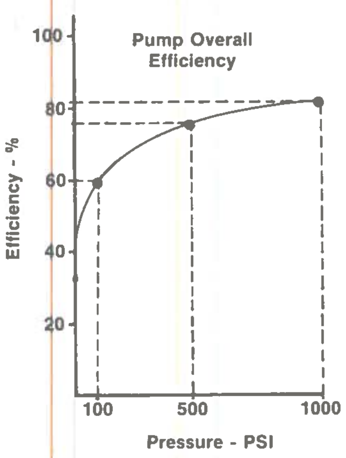
(Figure 4-26)



(Figure 4-27)



(Figure 4-28)



(Figure 4-29)



General Accumulator Symbol

(Figure 4-30)

With a variable volume pump, as leakage increases, the pump frequently can be readjusted to deliver the required flow rate. But, because of increased leakage, the pump will operate notter.

As illustrated, pump wear and its associated leakage affect no-flow compensator pressure. If the pressure under no-flow conditions is important to system performance, if it is being used for holding or clamping, the compensator setting can be readjusted. This can be continued until the compensator has been adjusted to its maximum value at which point compensator pressure will decrease as leakage increases.

UNLOADING PRESSURE COMPENSATED PUMP

During machine idle time, pressure compensated pumps are frequently not unloaded. As indicated earlier, generated power is reduced as the compensator setting of the pump is approached. The amount of input power during this period is generally quite low and is indicated by the input horsepower curve.

The illustrated performance curve points out that the input power while the pump is fully compensated at 500 PSI, is 1 HP. At a compensated pressure of 1000 PSI, input power is 1.5 HP.

In some systems, a pressure compensated pump is not allowed to compensate during idle time. This is the case when input power from an electric motor is less when the pump is unloaded than when compensated.

Referring to the pump performance curve again, we find that pump flow at full displacement is 10 GPM. Unloading this flow rate to tank at a pressure of 100 PSI would equal .583 HP (10 GPM x 100 PSI x .000583). Since electric motor input power equals hydraulic power divided by pump efficiency, input HP is approximately 1 HP when assuming an overall efficiency of 60% at 100 PSI (.583 / .6 = .97). this means if the pump is to be compensated at anything above 500 PSI (1 HP input), it is wiser to unload the pump. This is especially true when it is considered that the input HP curve shows pump performance under new conditions and does not take into account wear.

Pressure compensated pumps may or may not be compensated during machine idle time.

ACCUMULATOR

An accumulator is an energy storage device. It can be considered analogous to a battery, which stores

a current to be used upon command. An accumulator holds a volume of fluid at a specific pressure.

Different types of hydraulic accumulators are used in fluid power systems. The most common construction uses an inert gas in a bag or acting on a piston. Fluid enters the accumulator and is opposed by bag or piston and pressure in turn rises.

This type of energy storage device can be used as a supplement to the pump's supply, where short duration high flow demands are needed. It can also be used to supply a pressure for holding or pressing. In this type of application, the accumulator maintains system pressure while the pump may be unloaded.

There are five important points to consider when dealing with the accumulator. They are:

1. Minimum system pressure
2. Maximum pressure available
3. Precharge pressure
4. Isothermal or adiabatic charging or discharging
5. Method of unloading or shut down

These above points may be expanded as follows:

1. **Minimum System Pressure** — The minimum system pressure is determined by the lowest working pressure available for the condition to do work.
2. **Maximum Pressure** — Determined by the setting of the relief valve or by how much pressure is needed to achieve the needed volume from the accumulator.
3. **Precharge Pressure** — 100 psi (6.9 bar) less than the minimum system pressure. In this way (for piston accumulators) the piston never bottoms out and the gas pressure with relation to system pressure across the piston is always essentially equal to minimize gas or oil migration.

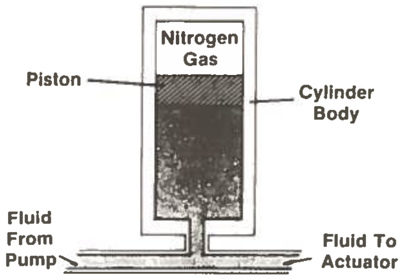
Precharge =
Minimum System Pressure - 100 PSI (6.9 bar)
(4-11)

ISOTHERMAL & ADIABETIC CHARGING

Since hydro-pneumatic accumulators use compressed gas to maintain pressure on a liquid, gas properties affect accumulator operation.

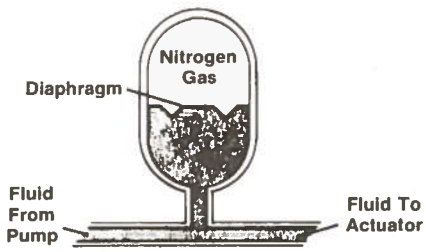
As a hydro-pneumatic accumulator is filled with liquid, gas is compressed. We know from Industrial Hydraulic Technology that as a gas is compressed it heats up. And, with pressure remaining constant, a heated gas occupies more space than a gas at a lower temperature.

Isothermal describes the operation of an accumula-



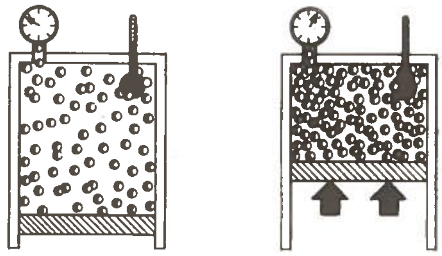
Piston Type Accumulator

(Figure 4-31)



Diaphragm Type Accumulator

(Figure 4-32)

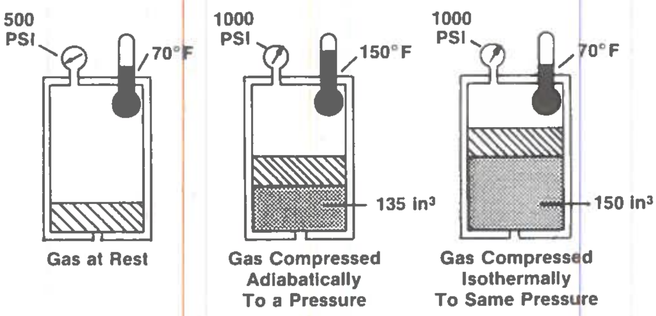


Gas Compression

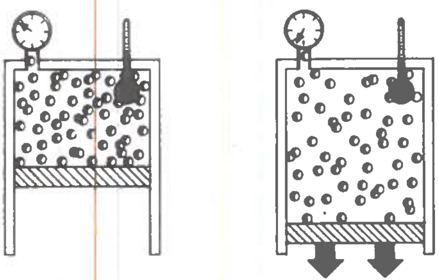
(Figure 4-33)

tor as the gas is maintained at a constant temperature. While an accumulator is being filled, isothermal operation indicates that the gas is being compressed slowly enough for the heat of compression to dissipate.

Adiabatic describes the operation of an accumulator as gas temperature changes. While an accumulator is being filled, adiabatic operation indicates that the gas is being compressed rapidly so that all heat of compression is retained.



(Figure 4-34)



Gas Expansion

(Figure 4-35)

A hydro-pneumatic accumulator which is being charged with liquid up to a certain pressure, will hold more liquid if it is charged isothermally rather than adiabatically.

Illustrated is a piston accumulator void of liquid. A pressure gage at accumulator top indicates a gas pressure of 500 PSI; a thermometer indicates a temperature of 70° F. Assume that the accumulator is going to be filled with liquid until a pressure of 1000 PSI is reached. Fluid ceases to enter the accumulator at this point because this is the maximum pressure allowed to develop by pump.

As the accumulator is charged adiabatically, pressure and temperature begin to climb. When pressure reaches 1000 PSI, fluid ceases to enter the accumulator inlet. At that point, the temperature is 150° F and the accumulator holds 133 in^3 of liquid.

Assume now that the accumulator is charged isothermally. Pressure begins to climb, but temperature remains the same. Charging takes place so slowly that heat of compression dissipates. When a pressure of 1000 PSI is reached, fluid ceases to enter accumulator inlet. At that point, temperature is still 70° F and the accumulator holds 150 in^3 of oil.

A hydro-pneumatic accumulator operated isothermally (slowly) will be charged with more liquid than if it were operated adiabatically (quickly).

ISOTHERMAL AND ADIABATIC DISCHARGING

As a hydro-pneumatic accumulator discharges liquid, gas expands. We found in Industrial Hydraulic Technology that as gas expands it cools. With pressure remaining constant, a cool gas occupies less space than a gas at an elevated temperature.

Isothermal and adiabatic describe the operation of an accumulator as it discharges fluid. An accumulator discharging fluid under isothermal conditions, indicates that discharge occurs slowly as gas expands; it is capable of acquiring heat from the ambient through accumulator walls or from the fluid. Adiabatic operation indicates that discharging

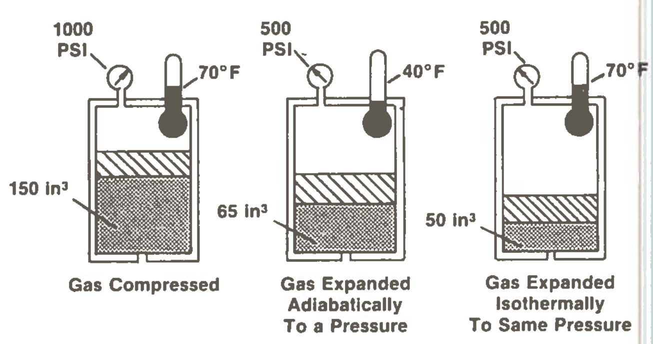
occurs rapidly with no heat gain; as gas expands it cools.

A hydro-pneumatic accumulator which is discharging liquid until a lower pressure is reached, will discharge more liquid if it is discharged isothermally rather than adiabatically.

Illustrated is a piston accumulator which is charged with liquid to a pressure of 1000 PSI. A thermometer at accumulator top indicates a gas temperature of 70°F. Assume that the accumulator holds 150 in³ of oil. Assume further that when a directional valve downstream is shifted, fluid will be discharged until accumulator pressure reaches 500 PSI. Fluid ceases to discharge from the accumulator at this time because 500 PSI is the working pressure of the system. This means a pressure differential no longer exists at this point to develop a discharge flow.

As the accumulator discharges adiabatically, gas pressure and temperature begin to drop. When gas pressure reaches 500 PSI, fluid ceases to exit the accumulator. At that point, temperature of the gas of 40°F and the accumulator holds 65 in³ of oil. 85 in³ have therefore been discharged.

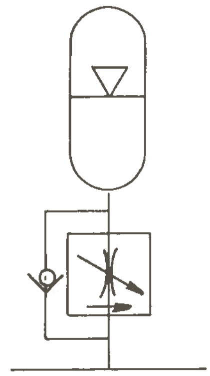
A hydro-pneumatic accumulator operated isothermally (Slowly) will discharge more liquid than if it were operated adiabatically (quickly). Ordinarily, accumulators are charged and discharged adiabatically.



(Figure 4-36)

CONTROL OF USABLE VOLUME DISCHARGE

The usable volume of an accumulator should be discharged at a controlled rate. If an accumulator is required to maintain system pressure, this controlled rate is automatically achieved by the leakage fluid it has to replace. However, an accumulator which is used to develop a pressurized flow can discharge its usable volume too rapidly as a downstream directional valve is shifted. For this reason, accumulators in this application are often equipped with a flow control and bypass check at their inlet-outlet port.



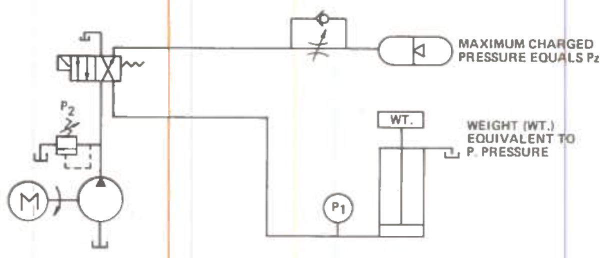
SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

(Figure 4-37)

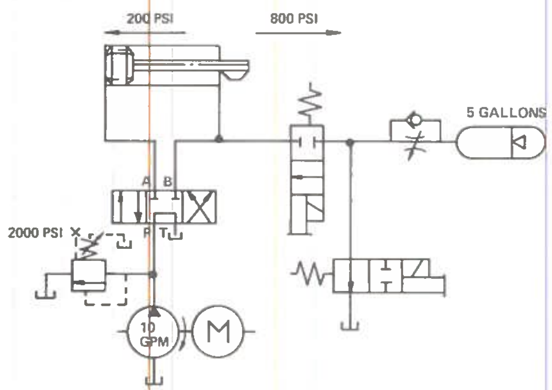
ACCUMULATOR CIRCUIT EFFICIENCY

It may be surprising but the circuit efficiency of an accumulator is not 100%. To develop the equation of accumulator efficiency; let's look at the following circuit: (Figure 4-38)

The pump will charge the accumulator a flow rate Q from a minimum system pressure of P_1 to a maximum pressure of P_2.



(Figure 4-38)



(Figure 4-39)

The power out to the system would be:

$$P \text{ out} = Q \times P_1 \times K$$

(4-5)

The power into the accumulator would be proportional to the pump flow and the average pressure. Exactly, it would be:

$$P \text{ in} = Q \times \frac{(P1 + P2)}{2} \times K$$

(4-12)

Since the efficiency of any system is power out compared to the power in; we may say:

$$\text{Efficiency} = \frac{P \text{ out}}{P \text{ in}} = \frac{Q \times P1 \times K}{Q \times \frac{(P1 + P2)}{2} \times K} \times 100$$

OR

$$\text{Efficiency} = \frac{2P1}{P1 + P2} \times 100$$

(4-13)

Good engineering practice would be to keep the efficiency around 50%.

Let us now work out an example of an accumulator problem.

EXAMPLE: (4-5)

An accumulator circuit is given as shown. The accumulator is used to supply oil in the return stroke mode. The pump has enough idle time to charge the accumulator to 2000 PSI. (Figure 4-39)

a. What is the precharge?
b. What is the volume of oil expelled?
c. What is the minimum idle time?
d. What HP does it take to charge the accumulator?
e. The cycle efficiency?

The charging and discharging is less than three minutes.

SOLUTION:

a. The precharge is 100 psi less than the minimum working pressure. In this case, the minimum working pressure is 200 PSI, therefore:

Minimum working pressure — 100 PSI = precharge
100 PSI = precharge pressure

b. The chart for the particular accumulator in this system is found in the catalog. It is shown in (Figure 4-40). The numbers at the left are the precharge pressure. The operating pressures (minimum & maximum pressures) are given at the top. Adiabatic values are given in bold print and isothermal values are in light print. In this

case, we have adiabatic operation therfore; the bold print will be used.

Entering the chart at the precharge pressure and following the numbers to the right, the volume under the 200 psi operating pressure and the volume under the 2000 psi pressure is recorded. They are:

@200 psi pressure there are 427 cubic inches of oil stored.
@2000 psi pressure there are 1035 cubic inches of oil stored.

The oil discharge will be:

Oil Discharge =
oil stored at high pressure - oil at low pressure

Oil Discharge = $1033 in^3 - 427 in^2$

Oil Discharge = $608 in^3$ of oil which may be discharged between these two pressure levels.

c. The idle time needed to charge this accumulator is dependent upon the delivery of the pump. We have tried to find out how long it takes to pump, to supply the discharge volume. In this case, 608 cubic inches of oil must be provided by a 10 GPM pump. We need to know how much flow 10 GPM will deliver the one second in cubic inches.

$$\text{Flow Rate } (in^3/sec) = \frac{\text{GPM} \times 231\ in^3}{\text{Gal.}} \times \frac{1\ \text{min.}}{60\ \text{sec.}} =$$

$$\text{GPM} \times 3.85$$

(4-15)

Flow Rate = 10 GPM x 3.85 = 38.5 in^3/sec.

Dividing the flow needed by the flow delivered, we will find the idle time needed to charge the accumulator.

$$\text{Idle Time (sec)} = \frac{\text{Flow discharged from accumulator } (in^3)}{\text{Flow rate delivered by pump(GPM)} \times 3.85}$$

(4-16)

$$\text{Idle Time} = \frac{608}{38.5} = 15.7 \text{ sec.}$$

15.7 seconds of idle time is needed to charge the accumulator.

d. The horsepower necessary to charge the accumulator can be determined by the horsepower equation given earlier. (4-12)

Substituting the values into formula (4-12) we calculate:

$$HP = GPM \times PSI \times .000583 =$$

$$HP = GPM \times \frac{(P1 + P2) \times .000583}{2}$$

$$= 10\ GPM \times 1100\ PSI \times .000583$$

$$= 6.413\ HP$$

e. The efficiency of the accumulator circuit can be found by employing (4-13)

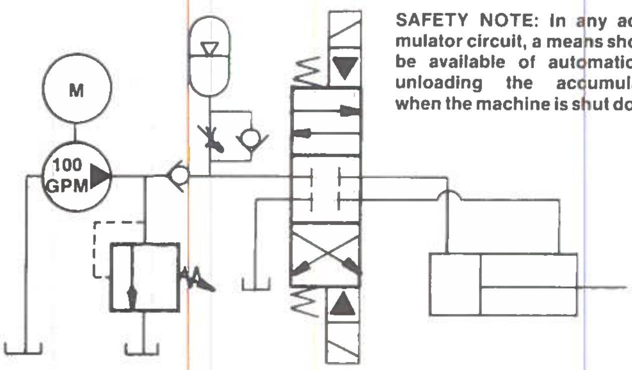
$$\text{Efficiency} = \frac{2P1}{P1 + P2} \times 100 = \frac{2(200)}{200 + 2000} \times 100 =$$

$$\frac{400}{2200} \times 100 = 18\%$$

GAS PRECHARGE PRESSURE — PSI (gage)	OPERATING PRESSURE — PSI (gage)														
	100	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
100		**427**	610	711	779	827	866	895	920	937	949	968	979	990	1000
		554	*757*	*860*	*924*	*967*	*1000*	*1020*	*1040*	*1057*	*1065*	*1079*	*1086*	*1093*	*1101*

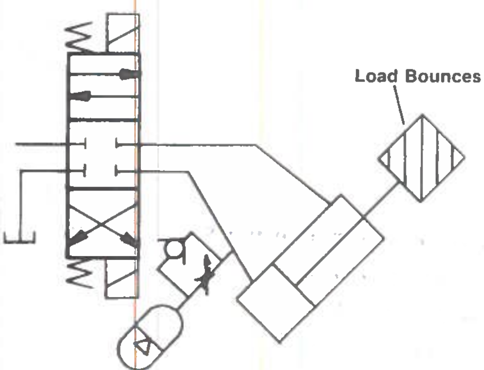
GAS PRECHARGE PRESSURE — PSI (gage)	OPERATING PRESSURE — PSI (gage)														
	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900	3000
100	1009	1015	1021	1029	**1035**	1039	1044	1050	1056	1058	1061	1063	1066	1070	1072
	1107	*1111*	*1118*	*1121*	*1124*	*1128*	*1131*	*1133*	*1135*	*1138*	*1140*	*1141*	*1142*	*1143*	*1144*

(Figure 4-40)



SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

(Figure 4-41)



SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

(Figure 4-42)

ACCUMULATORS USED FOR ABSORBING SHOCK

Hydro-pneumatic accumulators are sometimes used to absorb system shock even through in this application they are difficult to properly design into a system.

Shock in a hydraulic system may be developed from the inertia of a load attached to a cylinder or motor. Or, it may be caused by fluid inertia when system flow is suddenly blocked or changes direction as a directional valve is shifted quickly. An accumulator in the circuit will absorb some of the shock and not allow it to be transmitted fully throughout the system.

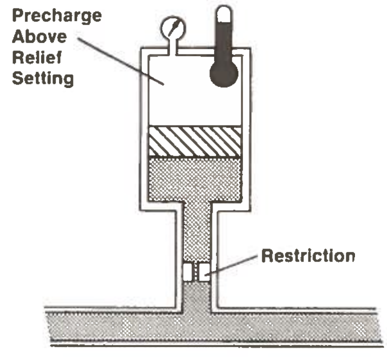
In the circuit illustrated, a pump is delivering 100 GPM to a cylinder at the required working pressure. If the closed center directional valve upstream from the cylinder is centered while work is occurring, the pressurized flow of 100 GPM will be stopped all at once resulting in hydraulic shock reverberating in the system. An accumulator positioned ahead of the directional valve absorbs and reduces the shock effects.

Shock may also occur in a hydraulic system due to external mechanical forces. In the circuit illustrated, the load attached to the cylinder has a tendency to bounce causing the rod to be pushed in and shock generated. An accumulator positioned in the cylinder line can help reduce the shock effects.

PRECHARGE AFFECTS SHOCK ABSORBER OPERATION

Precharge of a hydro-pneumatic accumulator affects its operation as a shock absorber.

Shock generation in a hydraulic system is the result of fast pressure rises due to an external mechanical force acting on a cylinder or hydraulic motor, or the result of liquid crashing into a component as a valve is suddenly closed. An accumulator acts to reduce a shock effect by limiting pressure rise.



(Figure 4-43)

In a hydraulic system, as shock pressures develop, they attempt to displace or push the fluid to another part of the system. But, since liquid is relatively incompressible, it won't move or compress.

Without an accumulator in the line, shock pressures can climb to a high value because they have a relatively solid base on which to build. With an accumulator in the line, the base for shock pressure development becomes soft.

Above a certain system pressure as a shock pressure begins to build, an accumulator absorbs the volume of liquid, the shock attempts to compress or displace. The line in which the accumulator is located becomes compressible above a certain point.

Gas precharge for a hydro-pneumatic accumulator used as a shock absorber is generally set slightly above the maximum working pressure of the line in which it is located. If the maximum pressure happens to be determined by the relief valve setting, gas precharge might be 100 PSI above this.

For example, if a relief valve were set for a maximum pressure of 2000 PSI, accumulator precharge would be 2100 PSI. And, 58 in³ accumulator was required to absorb 4 in³ of oil in order to dissipate the pressure. From the illustrated chart, it can be seen that a 58 in³ accumulator operating adiabatically with a precharge of 2100 PSI will allow pressure to climb to 2300 PSI as it absorbs 4 in³ of oil.

GAS PRECHARGE PRESSURE — PSI (gage)	OPERATING PRESSURE — PSI (gage)																				
	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900	3000
1000		4.4 6.1	8.1 11.2	11.4 15.5	14.3 19.2	16.8 22.4	19.1 25.3	21.2 27.8	23.0 29.9	24.7 32.0	26.2 33.7	27.6 35.4	28.9 36.8	30.2 38.2	31.2 39.4	32.3 40.5	33.2 41.6	34.2 42.6	34.9 43.4	35.8 44.3	36.5 45.7
1100			4.0 5.6	7.6 10.3	10.5 14.4	13.3 18.0	15.8 21.1	17.9 23.8	19.9 26.2	21.7 28.4	23.3 30.4	24.9 32.2	26.2 33.7	27.5 35.2	28.6 36.6	29.8 37.8	30.8 39.0	31.8 40.1	32.8 41.1	33.7 41.9	34.4 42.8
1200				3.7 5.2	7.0 9.7	9.9 13.5	12.5 16.9	14.7 19.8	16.9 22.5	18.8 24.9	20.5 27.0	22.2 29.0	23.6 30.7	25.0 32.4	26.2 33.7	27.4 35.1	28.5 36.4	29.7 37.7	30.7 38.8	31.5 39.7	32.3 40.5
1300					3.5 4.8	6.5 9.0	9.2 12.6	11.6 15.8	13.9 18.7	15.9 21.3	17.8 23.7	19.5 25.7	21.0 27.6	22.5 29.3	23.8 30.9	25.1 32.4	26.2 33.8	27.3 35.0	28.3 36.2	29.2 37.2	30.2 38.0
1400						3.3 4.5	6.1 8.4	8.7 11.9	11.0 14.9	13.1 17.7	15.1 20.3	16.9 22.6	18.6 24.6	20.1 26.4	21.4 28.1	22.8 29.7	24.0 31.2	25.1 32.5	26.2 33.8	27.3 34.9	28.2 36.0
1500							3.0 4.2	5.7 7.9	8.2 11.2	10.4 14.2	12.5 16.9	14.4 19.3	16.1 21.5	17.7 23.5	19.2 25.3	20.5 27.0	21.9 28.6	23.0 30.0	24.2 31.4	25.2 32.6	26.2 33.8
1600								2.9 4.0	5.4 7.5	7.8 10.7	9.9 13.5	11.8 16.0	13.6 18.4	15.2 20.5	16.9 22.5	18.3 24.3	19.6 26.0	21.0 27.6	22.2 28.9	23.3 30.3	24.3 31.7
1700									2.7 3.7	5.1 7.1	7.3 10.1	9.3 13.0	11.3 15.4	13.0 17.8	14.7 19.7	16.2 21.6	17.5 23.3	18.9 25.0	20.2 26.6	21.4 28.0	22.4 29.8
1800										2.5 3.5	4.9 6.8	7.0 9.7	9.0 12.2	10.8 14.8	12.5 16.9	14.0 18.9	15.5 20.8	16.9 22.6	18.2 24.2	19.3 25.6	20.5 27.1
1900											2.4 3.3	4.7 6.5	6.7 9.2	8.6 11.8	10.3 14.1	11.9 16.2	13.5 18.2	14.8 19.9	16.2 21.7	17.5 23.3	18.6 24.8
2000												2.3 3.2	4.5 6.2	6.4 8.8	8.3 11.4	9.9 13.5	11.6 15.6	13.0 17.5	14.7 19.2	15.6 20.9	16.9 22.5
2100													2.2 3.1	4.2 5.9	6.1 8.4	7.8 10.7	9.5 13.0	11.0 15.0	12.5 16.9	13.8 18.6	15.1 20.2
2200														2.1 3.0	4.1 5.6	5.9 8.2	7.6 10.4	9.2 12.5	10.7 14.5	12.0 16.3	13.4 18.1
2300															2.0 2.9	3.9 5.4	5.6 7.8	7.3 10.0	8.7 12.0	10.3 14.0	11.6 15.8
2400																2.0 2.7	3.7 5.2	5.4 7.5	7.0 9.7	8.5 11.7	9.9 13.5
2500																	1.8 2.6	3.6 5.0	5.2 7.3	6.8 9.4	8.3 11.4
2600																		1.8 2.5	3.5 4.8	5.0 6.9	6.5 9.0
2700																			1.7 2.4	3.4 4.7	4.8 6.7

Liquid Volume (in³) Stored in Accumulator

(Figure 4-44)

If the precharge were set too low, for instance, 1500 PSI, the accumulator would hold 14.4 in³ at 2100 PSI. In order to absorb 4 in³ pressure would climb to over 2300 PSI. As a matter of fact, pressure would climb to almost 2800 PSI before 4 in³ could be absorbed. Precharge of a hydro-pneumatic accumulator used as a shock absorber is quite important.

As an accumulator operates in a system as a shock absorber, it is generally required to get rid of the fluid it has accumulated in a controlled fashion. Commonly, accumulators in these applications are once again equipped with a restriction and bypass

check valve. With this arrangement, an accumulator can accept its required fluid, yet any fluid accumulation can bleed off through the restriction.

CONCLUSION

In this chapter, we covered the power generating devices of the fluid power system. There are two:

1. Pump
2. The Accumulator

The pump is a kinetic source of energy. It delivers a relatively constant flow rate at a particular pressure determined by the system. The equation necessary for pumps are:

SUCTION SIDE:
NPSH = 30 in. Hg. - min. inlet pressure
(4-3)
Max. vacuum at suction = barometric reading NPSH
(4-4)

PRESSURE SIDE:
Hp system = GPM x PSI x .000583
(4-5a)

$$P \text{ system} = (KW) \times \frac{LPM \times bar}{600} \times 100$$
(4-5b)

Input P = P motor = P system/efficiency of pump at system condition
(4-6)

Heat generation rate (P) = P motor – P pump
(4-7)

Heat generation rate (BTU) = (HP motor - HP system) x 2547
(4-8a)

Heat generation rate (J) = (P motor - P system) x .4147
(4-8b)

$$\text{Average Heat} = \frac{\sum_{n=1}^{\infty} Pn \times tn}{\sum_{n=1}^{\infty} tn}$$
(4-9)

Partial compensation = GPM x PSI x .000583 + Input HP at full comp.
(4-10a)

Partial compensation = LPM x bar ÷ 600 + Input P @ full comp.
(4-10b)

The accumulator is an energy storage device. It holds a particular volume of oil at pressure. It

releases its volume upon command, but then must be recharged.

Formulas used here are:

$$\text{Precharge Pressure} = \text{minimum system pressure} - 100 \text{ psi } (6.9 \text{ bar})$$

(4-11)

$$\text{Charging HP} = \text{pump flow(GPM)} \times \frac{(\text{max. pressure} + \text{min. pressure})}{2} \times .000583$$

(4-12a)

$$P = \text{LPM} \times \frac{(\text{Max pressure} + \text{min. pressure (bar)})}{2} \div 600$$

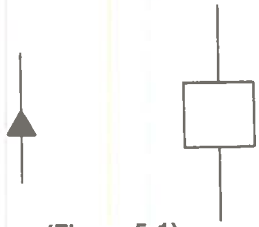
(4-12b)

$$\text{Cycle efficiency} = \frac{2P1}{P1 + P2} \times 100$$

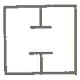
(4-13)

CHAPTER 5

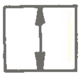
CHECK VALVES
DIRECTIONAL CONTROL VALVES



(Figure 5-1)



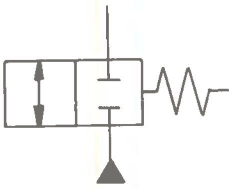
(Figure 5-2)



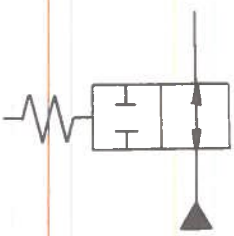
(Figure 5-3)



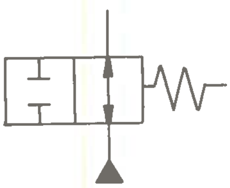
(Figure 5-4)



(Figure 5-5)



(Figure 5-6)



(Figure 5-7)

CONTROL VALVES AND THEIR SYMBOLOGY

All valves employed in a hydraulic circuit create a pressure loss and thus cause pressure to drop. The degree of pressure drop determines the particular valve's efficient use in a circuit. The more efficient the valve, the smaller the uncontrolled pressure drop; the less efficient the valve, the greater the uncontrolled pressure drop.

Before we enter into the valves in particular, let us quickly review the pertinent symbology that will be developed.

In (Figure 5-1), a solid triangle is pictured. This depicts a hydraulic or liquid source and flow direction from it.

Also pictured is a discrete position envelope of a valve with connecting lines. If the valve was a two-position valve, there would be two envelopes; three positon, three envelopes, etc. Taking this discrete position envelope and adding the appropriate symbols, the figure will become the blocked port or closed valve symbol. (Figure 5-2)

This means that fluid cannot flow between the connecting lines. Rearranging the symbol by inserting an arrow in the envelope, it becomes a valve that passes flow in both directions. (Figure 5-3)

If a doubled arrow is designated, this means that flow may exist in either direction. This symbol is restricted for use in this manual, when the pressure drop through the valve is less than 75 psi at rated flow capacity. If the pressure drop through a valve is greater than 75 psi, the following symbol will be employed. (Figure 5-4)

In this way, the actual circuit affect of valves may be more easily understood.

If the symbol developed in (Figure 5-3) is placed to the left of symbol (5-2) a two-way, two-position valve may be represented (Figure 5-5)

At this moment in time, no flow can exist. However, if the valve is shifted, the second flow conditions will come into effect. The second flow condition is one of passing flow. (Figure 5-6)

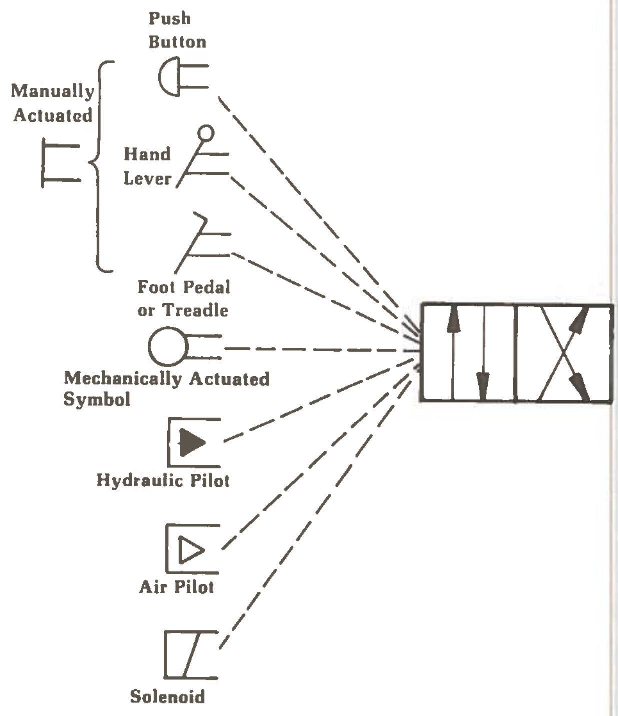
The valve pictured in (Figure 5-5) is a normally closed valve. In other words, no flow may pass through the valve in its initial conditions. If a normally open or passing valve was needed, the symbol shown in (Figure 5-7) would be drawn.

ACTUATORS

Previously, valve shifting was discussed, but the means by which this took place was not. Different

symbols have been developed to designate different types of shifting means.

The following chart illustrates the symbols commonly used. A more complete set can be found in the U.S. Standard ANSI Y 32.10.



FLOW LINES

There are three basic categories of lines that are found on a hydraulic schematic. The main flow lines, pressure and return, are drawn as solid lines. (Figure 5-8)

(Figure 5-8)

The width of the line does not alter the meaning. Henceforth, a solid line may mean ⅛" diameter tubing or 14" diameter pipe.

Dotted lines are used to indicate drain lines. (Figure 5-9)

(Figure 5-9)

Fluid in these lines is usually kept below 25 psi and the flow rates are generally small. It should be kept in mind that drain lines and return lines are not the same. Drain ports on components should have separate lines and not under any circumstances be connected to return lines.

Dashed lines indicate pilot paths. (Figure 5-10)

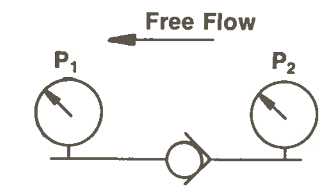
(Figure 5-10)

When we think of pilot paths, a low flow condition may be considered. This is due to the fact that flow rates are very small. The principal parameter being transferred is pressure. This means that pressure is the same at both ends of a pilot line, and that pressure changes are transferred virtually instantaneously.

Specific hydraulic valves, their symbols, and charcteristics will be covered in following chapters of this book.

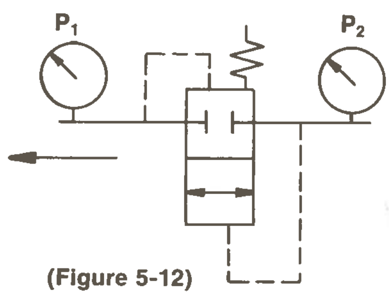
CHECK VALVES

A check valve is a two position, normally closed (not passing valve). Its simplified symbol takes on the following configuration. (Figure 5-11)



(Figure 5-11)

A composition symbol that describes the valve better would be (Figure 5-12)



(Figure 5-12)

As can be seen from (Figure 5-12), this valve has two pilot lines and a bias spring. The two pilot lines simultaneously monitor pressure on both sides of the valve. The spring insures that the valve is normally in its closed position and also determines how much greater P2 must be (compared to P1) to permit flow to pass through the valve. For instance, let the equivalent pressure of the spring be equal to 5 psi. This means that P2 must be 5 psi greater than P1 to allow fluid to flow. Anything less and no flow can

exist. In the "no flow" state, leakage is almost non-existant. In some applications, the spring force is specifically selected to impose or cause a specific pressure differential across the valve. For such applications, the spring force is usually greater than the normal bias provided by a normal spring and the spring for pressure equivalent is often referred to as its "cracking pressure".

Cracking pressure springs (springs which impose more pressure drop than the minimum required for good seating of the check) are available in different values, depending on circuit requirements. Some of the more common values are 5, 20, 30, 65, and 75 psi.

HORSEPOWER LOSS

HP loss or heat generation caused by this type of valve comes from two sources, each of which cause pressure drop.

1. The bias spring pressure drop
2. The mass flow rate pressure drop which is **dependent** on the flow

Equation (4-5) may be employed and rearranged to read:

Heat generation rate = flow rate x total pressure drop x constant (5-1)

The pressure drop due to flow for a particular valve must be extracted from catalog data.

EXAMPLE: (5-1)

Check valve 7B with a 20 psi cracking pressure is passing 8 GPM. What is the heat generation rate in horsepower?

SOLUTION:

Equation (5-1) is employed. However, the pressure drop due to flow must be found from the catalog.

Flow chart factors @ 5 psi cracking.

Size	Cv	Max. Flow	*P @ Max. Flow
7B	2.12	8 GPM	20 PSI

*For 65 psi cracking pressure add 65 to P values. For 20 psi cracking pressure add 20 To P valve. Specific gravity of oil used is .871.

From the above chart, the flow pressure drop is 20 psi.

Substituting into (5-1) and rearranging; we obtain:

Heat Generation Rate =
GPM x [bias spring (psi) + Flow pressure drop] x .000583

$$\text{Heat Generation Rate} = 8\ \text{GPM} \times \left[20\ \text{psi} + 20\ \text{psi}\right] \times .000583$$

$$\text{Heat Generation Rate} = .186\ \text{HP}$$

This example shows how the pressure drop was obtained through the check valve when it was operated at its maximum flow rate. However, if an intermediate rate was distributed through the valve, the pressure drop could not be read directly.

To obtain intermediate pressure drops, the following procedure should be used:

1. Determine the Cv if it is not already given.

$$Cv = \text{Flow}\sqrt{\frac{\text{specific gravity}}{\Delta P\ @\ \text{Flow}}} = Q\sqrt{\frac{Sg}{\Delta P}}$$

(5-2)

Cv is defined as the number of GPM of water a valve will pass with a 1 PSI drop across the valve.

Since the Cv is for water, we must convert it to a corrected Cv for the oil we are using.

First, the Reynolds number must be found according to formula (5-3).

$$NR = 17{,}250\ \frac{Q\ (GPM)}{\sqrt{Cv \times \nu}}$$

(5-3)

Where NR = Reynolds Number

Q = Flow rate

ν = Kinematic Viscosity - centistokes

Many times, viscosity is not given in terms of centistrokes. If this is the case, (Figure 5-13) must be consulted. Here we can easily convert SSU to centistokes. Centistokes are read directly opposite SSU.

With the information, (Figure 5-14) may be entered at the bottom with the Reynold's number and the Cv correction factor found. Equation (5-4) may now be used.

$$Cv\ \text{corrected} = \frac{Cv}{Fv}$$

(5-4)

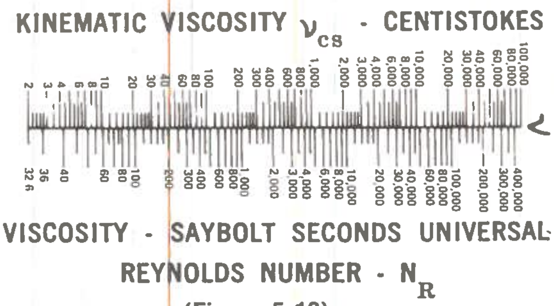
With this corrected Cv, the correct pressure drop can be calculated by rearranging (5-2)

EXAMPLE: (5-2)

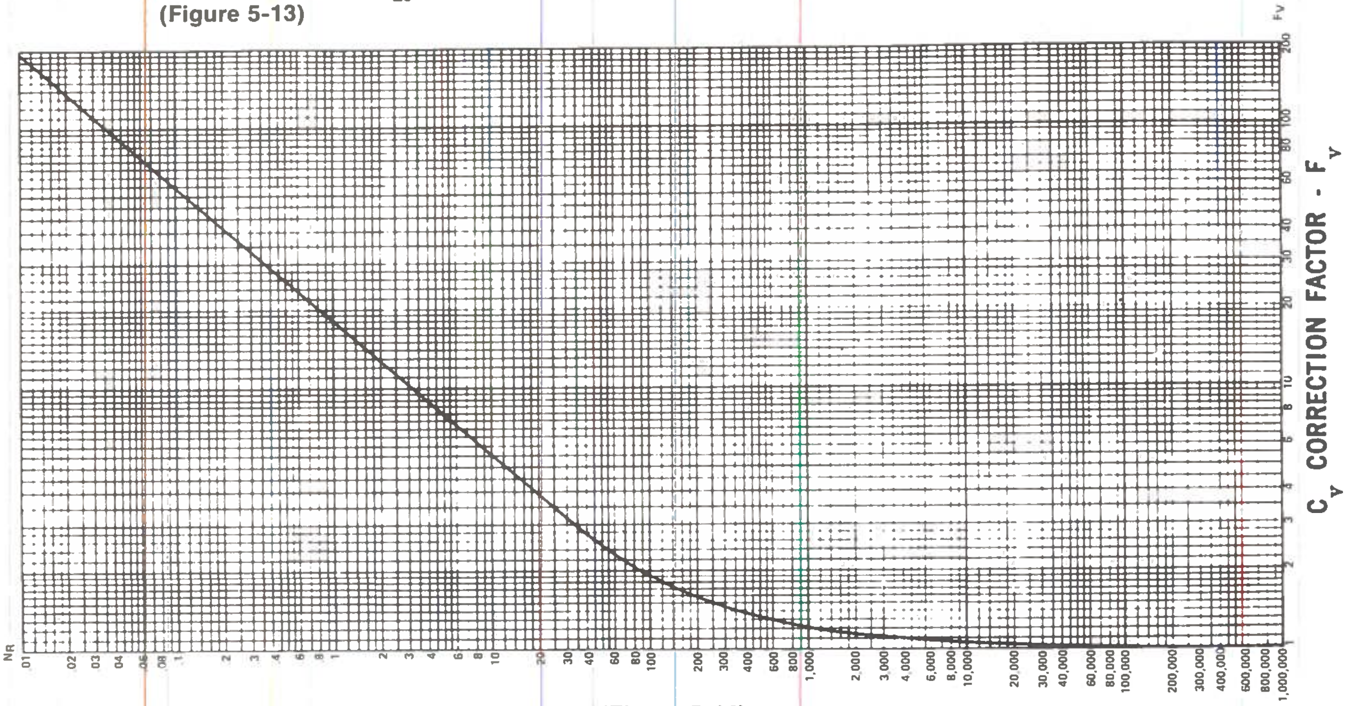
For the 5c check valve determine the pressure drop and heat generated for a 4 GPM flow. Oil being used has a specific gravity of 0.931 with a viscosity of 7000 SSU. Cv for the valve is 1.67 (obtained from catalog).

SOLUTION:

Cv is given. It is 1.67



(Figure 5-13)



(Figure 5-14)

The Reynold's number must be calculated next. To do this, we must know the viscosity in centistokes. (Figure 5-13) is used to find this.

SSU = 7000 = 1500 Centistokes

Calculation of the Reynold's number comes next.

$$NR = 17{,}250 \times \sqrt{\frac{4}{1.67 \times 1500}} = 35.6$$

Entering (5-14) at the bottom and moving upward to the curve and to the right, we find Fv equal to 3

Using equation (5-4)

$$\text{Cv corrected} = \frac{1.67}{3} = .5\,6$$

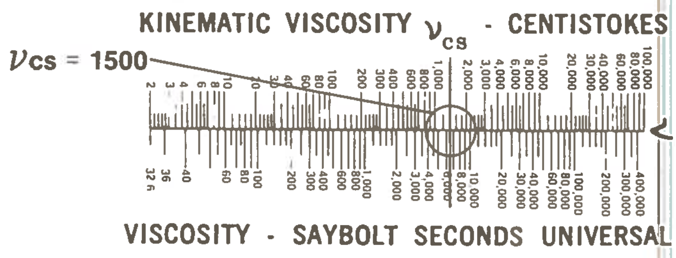
The pressure drop is now calculated using equation (5-2). Rearranging

$$\Delta P = \frac{(Q^2)}{(Cv^2)} \times Sg. = \frac{4^2}{.56^2} \times 0.931$$

ΔP = 48 psi drop

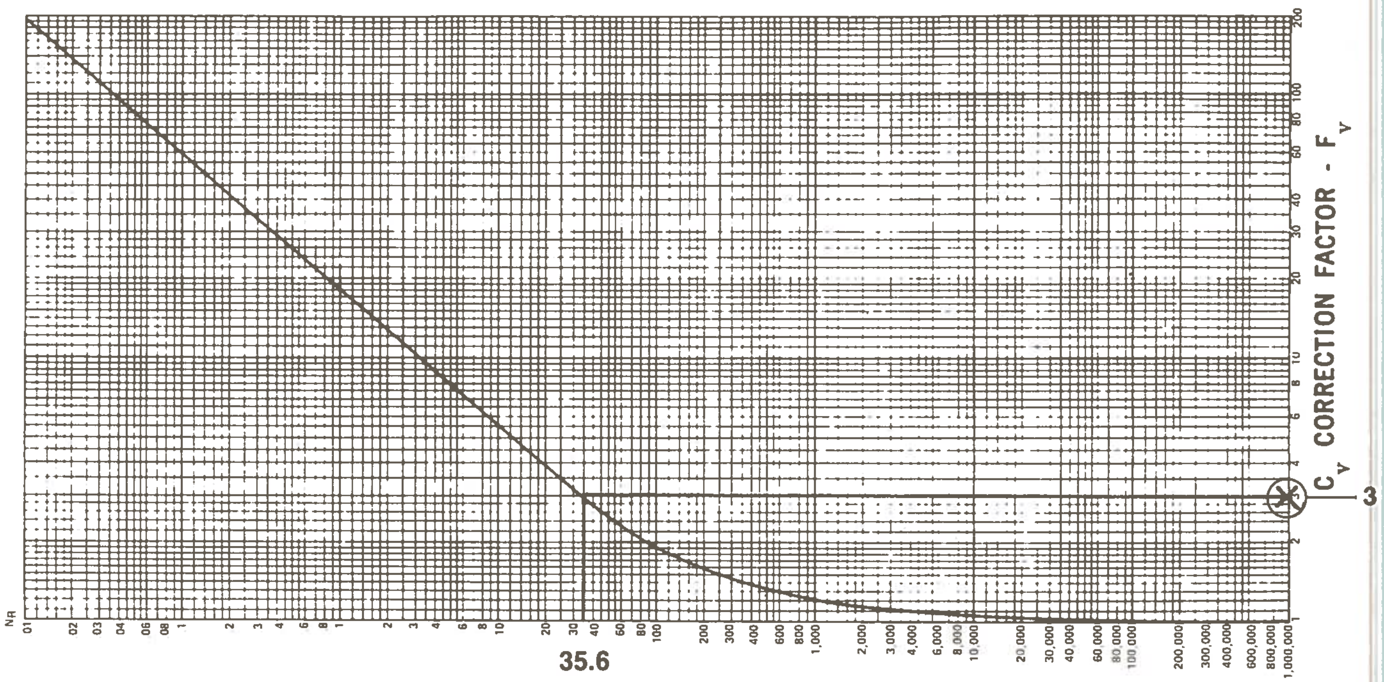
The heat generation rate is:

HP = GPM x ΔP x 0.000583



REYNOLDS NUMBER - N_R

(Figure 5-13)



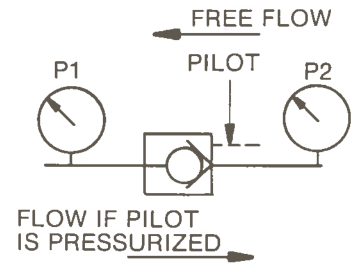
(Figure 5-14)

HP = 4 x 48 x 0.000583

HP = .111 HP in heat

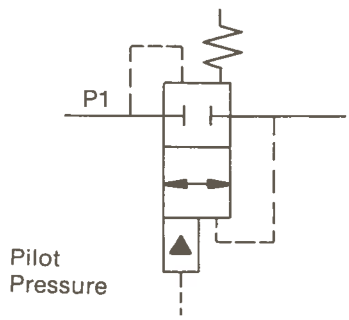
PILOT OPERATED CHECK VALVE

This type of check valve has an external connection which may be used to unseat the poppet so that flow may exist in the reverse condition. The simplified symbol is shown in (Figure 5-15).



(Figure 5-15)

The composite symbol for the above valve is (Figure 5-16).



(Figure 5-16)

When the external pilot is not pressurized, the valve is equivalent to a check valve. Pressure on both sides of the valve is monitored and the corresponding state taken. Now, let us consider the use of the hydraulic pilot.

The hydraulic pilot becomes important when flow is required in a reverse application. In order to have flow exist, the pressure at the pilot port must be at a specific pressure — say for example, 33% or ⅓ of P1 pressure, to shift the valve to a flow condition. If pilot pressure is lower than ⅓ P1, no flow exists. This ratio

3 to 1 (system pressure to pilot pressure) characterizes this particular type of valve. A typical name for this valve would be a 3 to 1 pilot operated check valve or a 3 to 1 P.O. check.

Note that many manufacturers offer pilot operated checks with ratios higher than 3 or 5 to 1. This type of valve may have a decompression poppet. This valve will open in the reverse direction if oil pressure could be decreased by causing a small bleed to exist at P1. If a small bleed will not cause P1 to drop, this type of valve may not operate at a high differential.

EXAMPLE: (5-3)

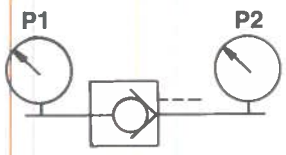
A 5 to 1 P.O. check is employed in the following systems. At what pressure will the valve open in:

a. Its free flow position?

b. Its no flow condition

P1 = 2000 psi (HINT: What is pilot pressure?)

Valve has a 5 psi cracking value. (Figure 5-17)



(Figure 5-17)

SOLUTION:

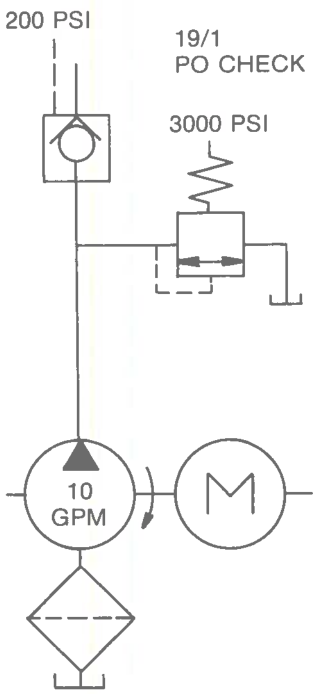
The valve is acting like a check and will open when P2 is 5 psi greater than P1 because the cracking pressure is 5 psi.

With pressure only at P1, the valve will open when the pilot port is actuated. The value is 5 to 1, therefore, only ⅕ of the pressure at P1 is needed to open the valve or:

Pilot pressure =
⅕ (2000) psi = 400 psi to open the valve.

EXAMPLE: (5-4)

The following circuit exists (Figure 5-18)



(Figure 5-18)

The machine operator pressurizes the pilot port of the check to start flow in the system. The pressure at the pilot is 200 psi.Nothing happens to the machine. What is wrong? The valve has a decompression poppet.

SOLUTION:

With P.O. checks with gains higher than 5 with decompression poppets, we must consider the criteria developed to see if the check will unseat. Since the pressure needed to unseat the valve is $1/19$ of the pressure at P1, the pilot pressure needed is:

Pilot Pressure = ratio x blocked pressure

Pilot Pressure = $1/19$ x (3000 psi) = 158 psi

Since 200 psi is being inputed, there should be enough pressure to unseat the poppet. The second criteria that must be met is: Will a small bleed at P1

cause the pressure of P1 to drop? The answer here is NO. The pump will keep P1 pressure high. Since the second criteria is not met, the valve will not open the main poppet in this application.

DIRECTIONAL CONTROL VALVES

Directional control valves usually control the direction of a cylinder. However, they may be used to perform other functions such as turn units off or on, unload pumps, etc. Different types of directional control valves will be covered here.

TWO POSITION VALVES

Two-position valves, as discussed earlier, have two discrete position envelopes. This means the valve has the ability to be positioned to either of two different flow configurations.

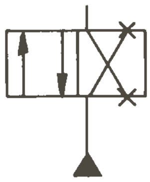
TWO WAY VALVES

Two way directional control valves have two main flow paths and usually provide either a "passing" or a "non passing" configuration or action. Thus, there is no flow in one condition and flow in the other. They may serve as safeties, interlocks, connectors to various system parts, logic or to prevent or allow flow of the working fluid. (Figure 5-19).



(Figure 5-19)

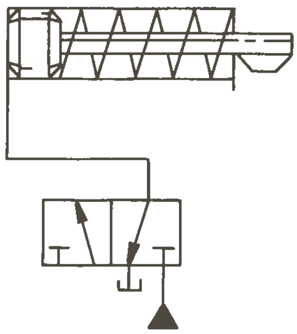
Because two way valves specifically designed for high pressure hydraulic service are not commonly available in larger sizes, a four-way valve may be substituted as a two way by plugging appropriate ports of the readily available four-way units. This is shown in (Figure 5-20)



(Figure 5-20)

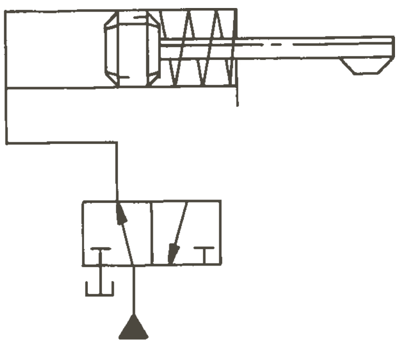
THREE-WAY VALVES

A typical function of this valve is to pressurize an actuator port in one position and exhaust the same actuator port in the other. This is quite useful in single acting actuators like rams and spring return cylinders. An example is shown in (Figure 5-21)



(Figure 5-21)

In this position, the spring is holding the cylinder retracted. Oil is returned to tank through the valve. When the valve is shifted, hydraulic power is directed to the cap end of the cylinder, causing it to extend. (Figure 5-22) These valves may also be used to select as shown in (Figure 5-23). This is a common use found in pneumatic systems.

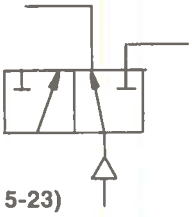


(Figure 5-22)

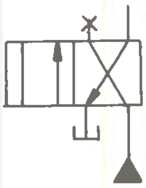
Like the two-way valve, three way valves are also often difficult to find in the hydraulic world.

However, a four-way valve can be used for three-way service in the following manner. (Figure 5-24)

FOUR WAY DIRECTIONAL CONTROL VALVES

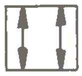


(Figure 5-23)



(Figure 5-24)

A very common function of a four-way directional valve is to cause the reciprocating action of a cylinder or bi-directional rotation of a motor. To perform this function, the spool directs flow from its pump passage (P) to one actuator port (A). The opposite cylinder port (B) is connected to the tank port (T). (Figure 5-25)



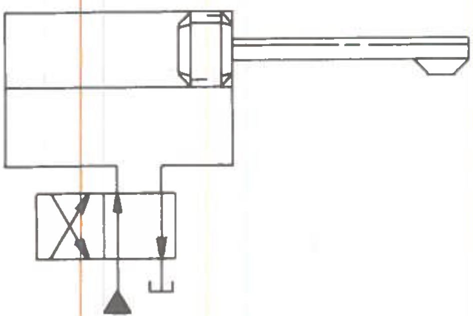
(Figure 5-25)

In the other position, the pump port (P) is connected to (B) and the (A) actuator port is connected to tank (T). (Figure 5-26)



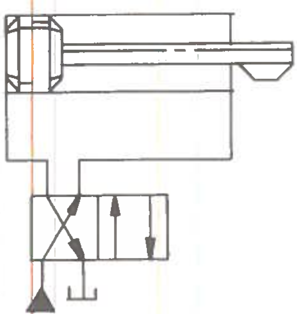
(Figure 5-26)

A typical example of the use of this valve is to cycle a double acting cylinder back and forth. In one position, the valve will allow flow to extend the cylinder. (Figure 5-27)



(Figure 5-27)

In the other position, the cylinder will retract. (Figure 5-28)



(Figure 5-28)

DIRECTIONAL VALVE ACTUATORS

As discussed previously, a two position direction control valve must be positioned in its positions by some means. The spool may be moved to these positions by mechanical, electrical, hydraulic, pneumatic or human energy, or a combination of some of these.

Manually operated valves are controlled by muscle. "Manual valves", valves whose alternate positions are a result of forces transmitted through pushbuttons, hand levers, foot pedals or treadles.

Mechanical operated valves are ones equipped with a roller actuated by some type of mechanical means, may be considered a limit valve. A limit valve tells you where you are, where you are not, or where you were.

Pilot operated valves are controls whose alternate positions are a result of fluid pressure forces which may be applied or removed. They are generally positioned by applied fluid pressure either air or hydraulic. Typically, pilot pressure is applied to the extreme spool lands to shift the spool.

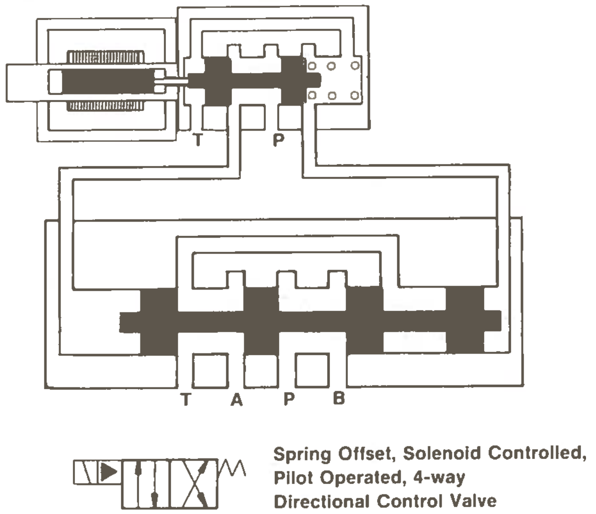
In the machine tool and non-mobile applications of hydraulics, solenoid operation is probably the most common. Here the electrical signal is converted into a linear force motion, which in turn moves the spool to one of its positions. When the solenoid acts directly on the spool, the operation of the valve is said to be "direct".

PILOT OPERATED SOLENOID VALVES

Because the force developed by the solenoid is

limited, the force available for shifting of large flow rate valves is limited. As a result, the directional valves which use solenoids directly to shift a spool are generally only less than 20 GPM for the nominal flow rate. Larger flowrate valves are operated by hydraulic pilot pressure controlled by the smaller valve mounted to the larger valve. Flow from the small valve is directed to the spool ends of the large valve in order that the valve spool may be shifted. These valves are designated as being "solenoid controlled-pilot operated" directional valves.

When dealing with pilot operated directional control valves actuated by internal pilots, a minimum pressure with respect to the tank port is required. A typical value for this pressure is 65 psi. Without at least a 65 psi differential, the valve may fail to shift properly. (Figure 5-29)



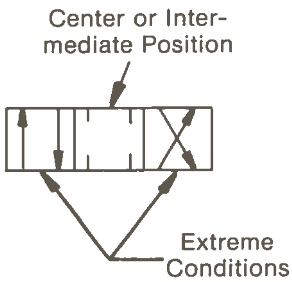
Spring Offset, Solenoid Controlled, Pilot Operated, 4-way Directional Control Valve

(Figure 5-29)

THREE POSITION VALVES

Four way directional control valves are normally available with two positions and also three position types. The two actuated flowpath conditons of a three position valve are similar to those shown for the two position valve. (Figure 5-29)

The neutral or intermediate condition may be used to perform needed logic required in an "at rest" signal condition. For this reason this position is often called the center condition of the valve. The two actuated or "extreme" conditions are directly dependent upon the control of the actuators.



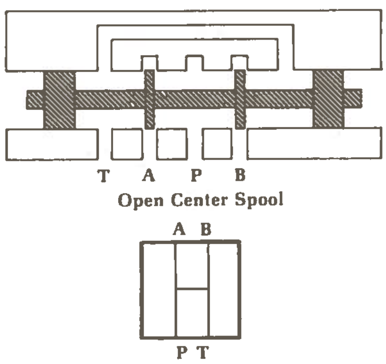
(Figure 5-30)

There are a variety of center conditions available with four-way valves. Some of the more popular ones are termed: "open center," "closed center," "tandem center" and "float center". These different center conditions are achieved through the use of different spools an provide an important control function in many hydraulic circuits. Center conditions generally affect the pressure drop of the fluid and on some configurations, the maximum flow rate that can pass through a valve without malfunction (resulting from flow forces). This will be covered further in following sections.

OPEN CENTER CONDITIONS

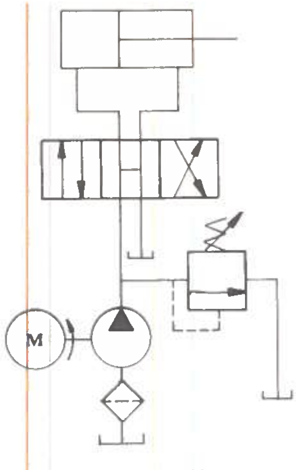
A directional valve with an "open center" condition has all ports — A, P, T, and B interconnected or open in the center or neutral position; thus the term open center. The configuration is as follows: (Figure (Figure 5-31)

This type of center allows relatively free movement of the actuator as well as unloads the pump. (Unloading a pump means to allow the pump flow to



(Figure 5-31)

return to tank under low pressure.) A circuit with the above characteristics would be as follows: (Figure 5-32)

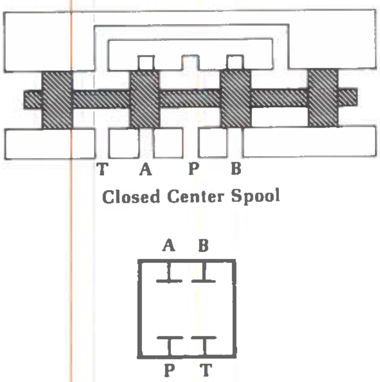


(Figure 5-32)

The disadvantages of an open center valve is that no other actuator may be operated once the valve is centered if open center valves are all used in parallel. Also, when using the open center condition in a solenoid controlled pilot operated valve, the necessary minimum pressure needed to shift the valve must be considered or the valve may malfunction.

CLOSED CENTER CONDITION

A directional valve with this condition has P, T, A, and B blocked or closed in the center condition. (Figure 5-33)



(Figure 5-33)

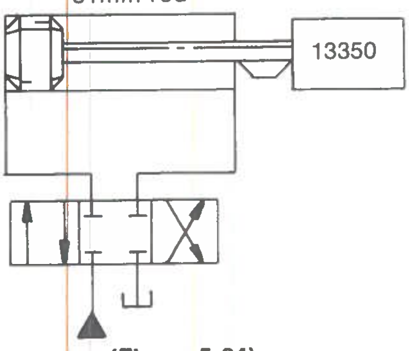
Because most valves depend on a closely fitted spool working in a bore for sealing, there is a small degree of leakage between all the ports. When the valve's in the center position, system pressure is present in the P passage of the valve. This pressure causes flow across the land to the A and B port which in turn causes flow to exist to tank. The pressures in the actuator passages at this point may increase to a particular value.

The value to which they increase has to do with the piston and annulus area as well as the load attached to the cylinder. The actual value of pressure is unimportant except for one case.

It should be checked to see if ½ the pressure existing at the P port when applied to the cylinder in a regenerative manner, will cause the piston to move. If it will, there may be a drifting out of the cylinder when the valve is centered.

EXAMPLE: (5-5)

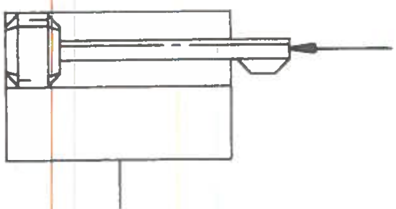
The circuit exists, as found in (Figure 5-34). Will the piston be able to drift when the valve is centered?



(Figure 5-34)

SOLUTION:

Consider the cylinder in a regenerative mode with ½ of the pressure of P at A and B. It becomes (Figure 5-35)


(Figure 5-35)

In a regenerative system, the effective area is the rod area, causing the working force to be:

$$F = (P \times Ar) / (0.1 \times 2)$$

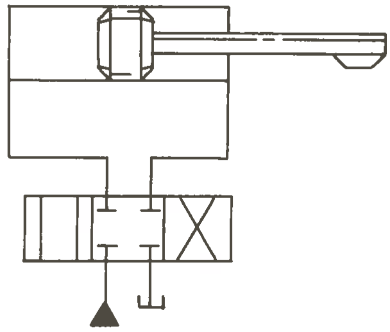
F = 17610 Newtons

Since this force would only have to overcome 13350 neutons, the cylinder may drift.

Another application for the closed center valve

would be holding a cylinder in a controlled position.

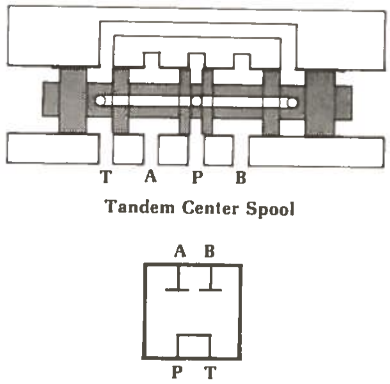
(Figure 5-36) illustrates a hydraulic cylinder connected to a three-position, four-port, four-way valve. When solenoid A is energized, the valve is shifted, allowing the oil to extend the cylinder. When voltage is removed, the cylinder will stay at its respective position. To return the cylinder, a voltage must be introduced at solenoid "B", shifting the valve to the left to reset the cylinder to initial conditions.



(Figure 5-36)

TANDEM CENTER VALVES

A "tandem center" condition stops the motion of the actuator while it simultaneously unloads the pump. (Figure 5-37)



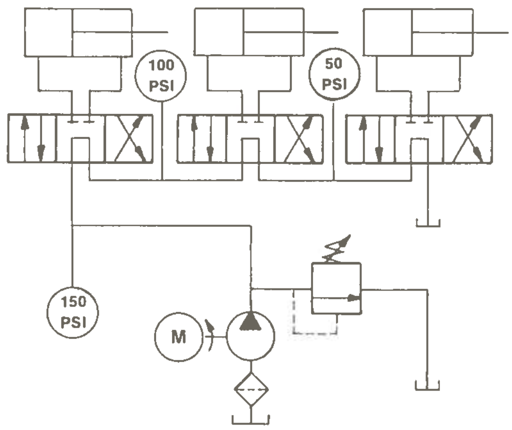
(Figure 5-37)

A directional valve with this center has the obvious advantage of unloading the pump. However, it has some disadvantages that are not apparent by looking at the symbol.

When a tandem center spool is used to achieve this center condition, the nominal flow rating of the valve is usually decreased. This reduction is necessary because of the large pressure drop present when flow exists from the supply port P to the tank port T.

In some designs, passages P to T are connected via a cored passage internal to the spool; since the flow enters and exhausts through holes in the spool, large pressure drops are often inherent.

It is not uncommon to see several tandem center valves connected in a series in a circuit. The idea behind this arrangement is that the actuators can act somewhat independently of each other and at the same time the pump can be unloaded when the tandem center valves are in the center position. But, since each valve may generate a 50 psi differential while at rated flow during center condition, system pressure at this point may not be as low as desired. (Figure 5-38)

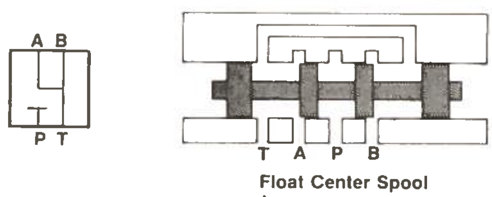


(Figure 5-38)

It should be noted that tandem center directional spool operate a little differently than other spools. Because of its construction, when the usual design tandem center spool is shifted toward the right side of the valve, flow passes from P to A. But, in any other spool, flow passes from P to B. Consequently, if a tandem center spool replaces another type of spool in an operating system, the actuator controlled by that directional valve may operate "backwards".

FLOAT CENTER VALVES

Float center valves block the pressure port (P) of the valve while simultaneously interconnecting the



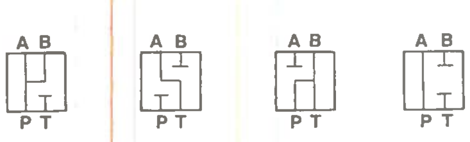
(Figure 5-39)

actuator ports (A and B) to the tank. (T). (Figure 5-39)

The float center valve is quite useful in applications where it is desired to be able to move the actuator by relatively small external forces when the valve is in the neutral or center position. The actuator is free to "float" thus giving the center its name.

OTHER CENTER CONDITIONS

Besides the center conditions already discussed, others exist. These even give greater flexibility for a system. Some what are offered are shown below:



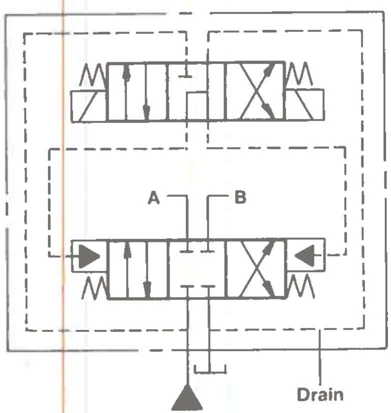
(Figure 5-40)

The versatility of three position valves are outstanding. Often one valve can control two separate and distinct actions as for example: Clutch or brake when properly applied.

DIRECTIONAL VALVE DRAIN

The pilot valve tank passage of a solenoid controlled, pilot operated directional valve is called a drain.

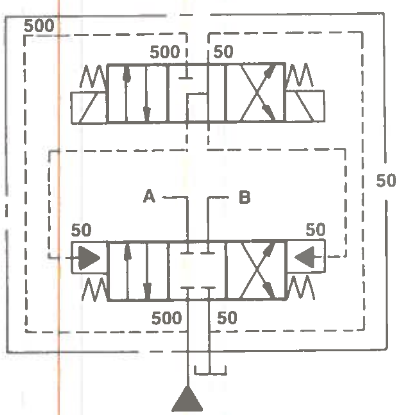
The composite symbol of a solenoid controlled, pilot operated directional valve shows that the tank port of the pilot valve is connected by means of an internal passage to the tank passage of the main valve. This internal passage connection is known as an internal drain. With an internal drain, pressure in the tank passage must be contended with in shifting the main valve spool. (Figure 5-41)



(Figure 5-41)

From the illustrated symbol, the P port of a closed center directional valve is subjected to 500 PSI system pressure. This pressure is also supplied to the solenoid pilot valve by means of an internal passage. Back pressure in the tank line is 50 PSI.

With the float center pilot valve centered, each end of the main valve spool is exposed to the back pressure. When the pilot valve is shifted, 500 PSI is directed to the left spool end. Inorder to shift the main spool to the right, pilot pressure must overcome 50 PSI back pressure and the spring which has a value of another 50 PSI. Of course, 500 PSI can accomplish this quite easily. The main spool shifts.



(Figure 5-42)

In some situations, the pilot valve is externally drained. This is the case when shock pressures are experienced in the tank line.

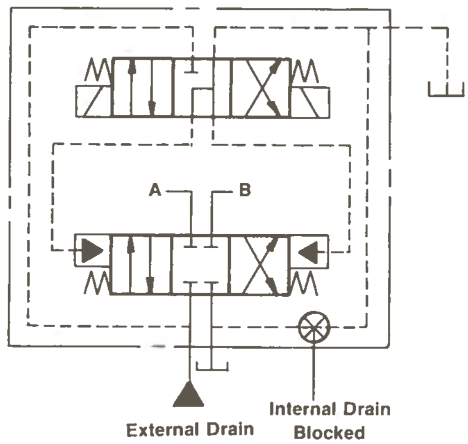
Assume that an internally drained directional valve has its spool held shifted to an extreme position with a pilot pressure of 500 PSI. This is the case in the last example. Keeping in mind that any pressure in the tank line acts on the main spool end drained to tank, assume now that a single acting cylinder discharges

in another part of the circuit. Pressure climbs in the tank line to 1000 PSI for a fraction of a second. Since the tank line is usually common for all valves, this pressure arrives at the main spool end drained to tank. With 500 PSI on one side and 1000 PSI on the other, the spool shifts in an undesired direction for a fraction of a second. This results in erratic motion of an actuator.

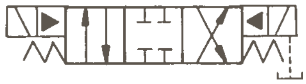
To remedy the situation, the valve should be externally drained. This is accomplished by removing the pilot valve from the main valve. This exposes the pilot valve porting on top of the main valve. The port pattern will consist of a "W" Porting arrangement. A and B ports, and two T ports. One of the T ports will be threaded. A pipe plug is inserted into this port blocking the internal drain. The pilot valve is replaced and a separate drain line as connected to the external drain port (Y port) of the main valve subplate. (Figure 5-43)

With a pilot operated directional valve externally drained, back pressure on a spool land is very low during shifting.

Externally draining a pilot operated directional valve is also required with tank line back pressure check valves. This will be illustrated below. (Figure 5-44)



(Figure 5-43)

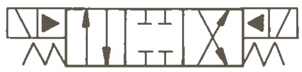


Simplified Symbol

(Figure 5-44)

DIRECTIONAL VALVE PILOT PRESSURE

Pressure supplied to the pilot valve of a solenoid controlled, pilot operated directional valve is known as pilot pressure. This is used to shift the main valve spool. (Figure 5-45)



Simplified Symbol

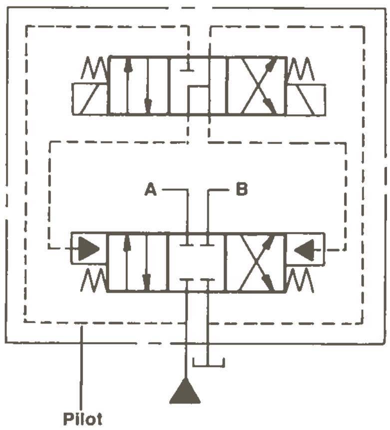
Solenoid Controlled, Pilot Operated Directional Valve

(Figure 5-45)

The composite symbol of a solenoid controlled, pilot operated directional valve shows that the P port of the pilot valve is connected by means of an internal passage to the P passage of the main valve. This is an internal pilot connection. (Figure 5-46)

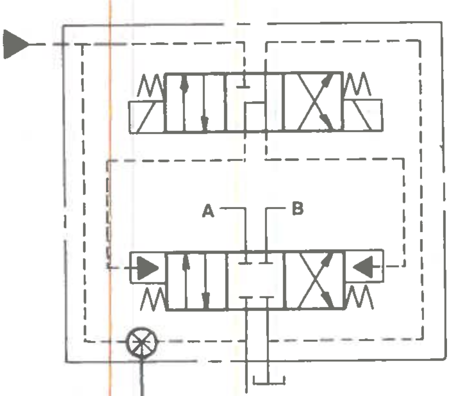
With an internal pilot, pilot pressure to shift the main valve spool has the same value as system pressure. From a previous example, a system pressure of 500 PSI was supplied to the main valve spool. This was used in overcoming spring and back pressure at a spool end during shifting. As long as the pilot pressure is sufficient to overcome spring and back pressure, the spool will shift.

In some situations, the valve is required to be externally piloted. This is the case when internal pilot pressure is either too low or too high. (Figure 5-47)



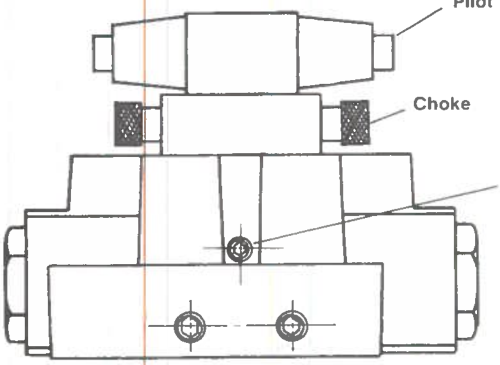
(Figure 5-46)

Assume that the system in which an internally piloted directional valve is operating, has erratic system pressure. In one instant system pressure

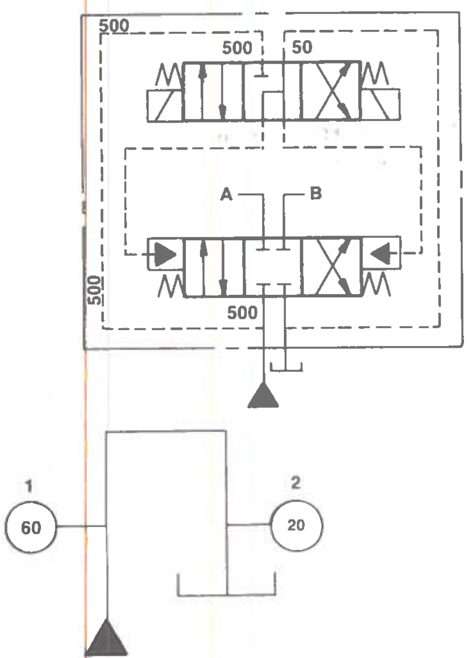




(Figure 5-47)



(Figure 5-48)



(Figure 5-49)

might be 500 PSI; the next instant it's 50 PSI. This situation cannot be depended on to shift the valve. To solve the problem, the valve can be externally piloted with an external, dependable source of pilot pressure.

Externally piloting a directional valve is also desireable if a system pressure is quite high. Assume that a system pressure of 2500 PSI is supplied internally to a pilot valve. When the main valve spool is required to be shifted from one extreme position to another, the pilot valve is shifted. 2500 PSI acts on a spool end, accelerating the spool quickly to an extreme. This may generate shock in the system.

Besides using choke controls, the pilot valve can be externally supplied with a pilot source of lower pressure. (Figure 5-48)

To change from internal pilot to external pilot, an internal pipe plug must be added. On the side of the main valve body, a pipe plug covers a port which communicates with the internal pilot passage. This plug is removed. Inside the port, the pilot passage and another threaded port can be seen. The internal port is plugged with a very small pipe plug (1/16 in). This blocks the internal pilot. The outer pipe plug is then replaced and the external pilot port (X port) on the main valve subplate is connected to the source of pilot pressure.

Externally piloting a pilot operated directional valve is sometimes required when a back pressure check valve is positioned in the pressure line ahead of the valve.

BACK PRESSURE CHECK VALVE

Pilot pressure to shift the main spool of a solenoid controlled, pilot operated directional valve frequently comes through the main valve body from the main system pressure.

In the illustrated symbol, a pilot operated directional valve with a closed center is shown. With 500 PSI present at the valve P port, 500 PSI is supplied to the pilot valve through an internal passage. To shift the main spool, the solenoid pilot valve is shifted, 500 PSI is directed to one end of the main spool and the spool shifts.

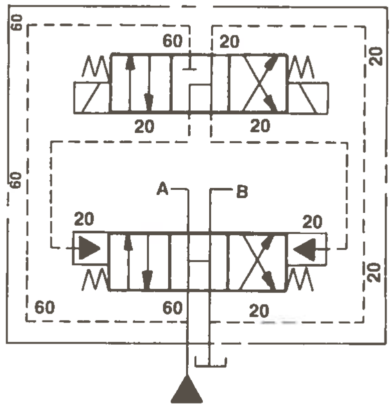
Solenoid controlled, pilot operated directional valves which have P connected to tank in the center position, do not have a readily available supply of pressure for the pilot valve. This is the case with open and tandem center directional valves.

In the illustration, pump flow returns directly to tank through system piping. Gages in the line indicate

that system pressure at gage 1 is 60 PSI; and at gage 2, pressure is 20 PSI. (Figure 5-49)

This exact situation exists with a tandem and open center directional valve while centered. Gage 1 would indicate the point within a valve where the internal pilot pressure is picked up. Gage 2 would indicate the point where the pilot drain is connected internally.

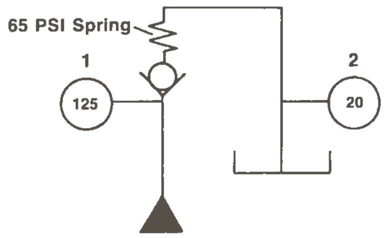
With this situation, 60 PSI is supplied to the main spool valve. This pressure must overcome 20 PSI back pressure and a 50 PSI spring. Of course, this cannot be done. (Figure 5-50)



(Figure 5-50)

To remedy the situation, the valve can be externally supplied with pilot pressure from a remote source. The pilot valve could also be externally drained, but in this instance eliminating 20 PSI back pressure may not ensure that the spool will shift.

Frequently, with tandem and open center pilot operated valves, back pressure check valves are used. A back pressure check has a stiffer spring than a normal check valve. This spring requires that more pressure be present at the valve inlet to push the poppet off its seat. The additional pressure is then supplied to the pilot valve. (Figure 5-51)

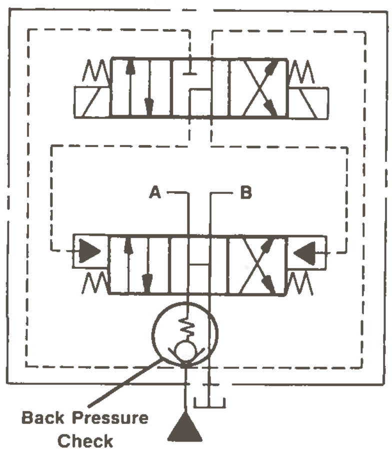


(Figure 5-51)

Back pressure check valves can be positioned in either pressure or tank lines. They can be located within the valve or external to the valve.

Some manufacturers offer a 65 PSI back pressure check valve within the P port of the main valve. Its effect can be illustrated by considering a previous example.

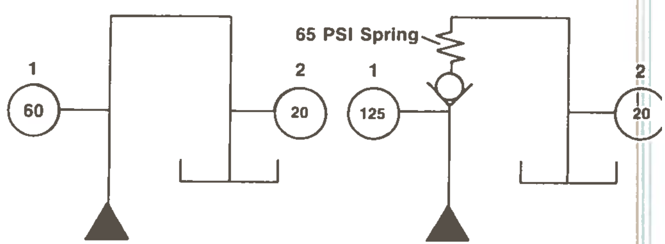
In the illustration, pump flow dumps directly to tank through system piping. Gage 1 indicates 60 PSI; Gage 2 indicates 20 PSI. Putting a 65 PSI check valve after the gage 1 point, requires that pressure to get back to tank be an additional 65 PSI for a 125 PSI total. If Gage 1 indicated the point where internal pilot pressure is picked up, then 125 PSI pilot pressure would be supplied to the pilot valve. When shifting is required, 125 PSI is directed to one spool and overcoming 20 PSI back pressure and a 50 PSI spring.



(Figure 5-52)

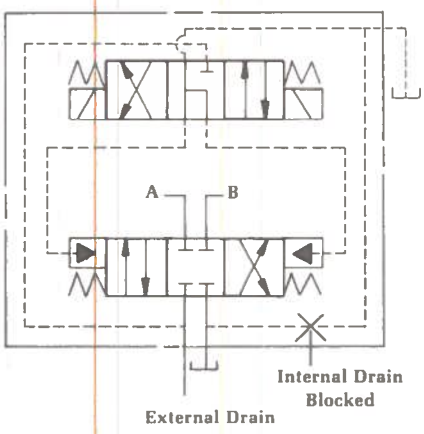
The back pressure check valve can also be external to the valve. In this instance, the pilot valve is externally piloted.

Other valve manufacturers offer a 65 PSI back pressure check valve in the tank port of the main valve. Its effect can be illustrated by the following example: (Figure 5-52)

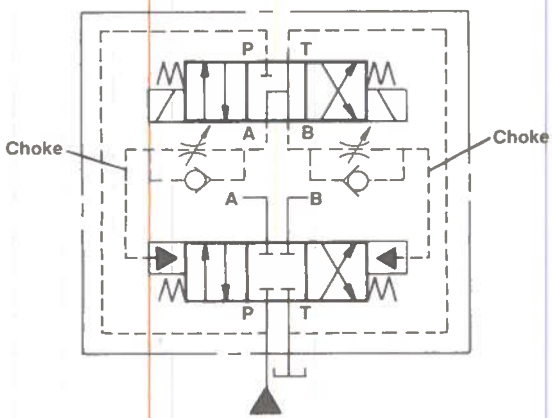


(Figure 5-53)

Once again, pump flow dumps directly to tank



(Figure 5-54)



(Figure 5-55)

through system piping. Gage 1 indicates 60 PSI; Gage 2 indicates 20 PSI. With a 65 PSI back pressure check placed after Gage 2, fluid pressure must be an additional 65 PSI or 85 PSI in order to get to tank. This also means gage 1 indicates an additional 65 PSI or 125 PSI. (Figure 5-53)

Assume that gage point 1 is where pilot pressure is picked up and gage 2 is the place where the internal drain is connected. With this situation, the pilot valve is supplied with 125 PSI pilot pressure. In order to shift the valve, this pressure must overcome a back pressure of 85 PSI and 50 PSI spring. The spool will not shift. To remedy the situation, the pilot valve is externally drained eliminating the 85 PSI back pressure.

A back pressure check valve may also be located externally in the tank line.

Any time a tank line back pressure check is used with tandem and open center valves, the valve is externally drained. (Figure 5-54)

CHOKE CONTROL

As the main spool of the pilot operated directional valve is shifted, shock can be developed as large fluid flows are forced to change direction quickly. A choke control slows the spool shift of the main valve so that shock is reduced. (Figure 5-55)

A choke control is a valve "sandwich" which fits in between a main valve body and pilot valve of a solenoid controlled, pilot operated directional valve. It consists of two needle valves and two bypass check valves. The valve sandwich is arranged so that as the spool shifts in one direction a needle valve meters flow out of the pilot-spring chamber; the other needle valve is bypassed. As the spool shifts in the opposite direction, the other needle valve meters flow out of the other pilot-spring chamber. The more the needle valves are adjusted in, the more flow is restricted or choked. Consequently, the slower is the spool shift.

A choke control does not eliminate shock due to spool shift; it reduces shock. If a crash is experienced each time a large directional valve is shifted, a choke control will reduce the crash to a bang. If the shock is a bang, chokes reduce it to a thump.

DIRECTIONAL VALVE SIZE

Four way, directional control valves used in industry are manufactured in many sizes. 1/4, 3/8, 1/2, 3/4, and 1 1/4. The maximum flow rating on these valves has typically been 3-5 GPM, 10-12 GPM, 20 GPM, 35 GPM and 100 GPM respectively. Although new design and manufacturing techniques are

increasing the rated flow capacities for all sizes, pressure drops vary depending on spool configuration and valve design. Technical data should be consulted to determine the normal pressure drop through a valve for any given spool configuration or flow rate.

With the trend to increase the nominal flow rate of a particular line of valves, the pipe size rating is no longer sufficient. New NFPA designation only refer to the valve interchange dimensions. For example, with this standard an individual is not tied to using ¾" pipe with a nominal ¾ valve. If this valve was rated for 80 GPM, (which is the trend), ¾" pipe would not be sufficiently large to keep the flow velocity below 15 ft/sec.

DETERMINING THE PRESSURE DROP THROUGH A VALVE

The pressure drop through a valve is proportional to the square of the velocity through the valve. The procedure for finding the pressure drop is as follows:

2-Position Valves

a. Determine the flow rates through the different paths in the valve.

b. With these flow rates enter the graphs displaying the particular valve characteristics and extract the pressure drop.

c. Correct for changes of viscosity.

3-Position Valves

a. Determine the type of center conditions on the particular valve in question.

b. Determine the flow rate through the different passages in all conditions.

c. With the flow rates enter the graphs of the particular valve and extract the pressure drop.

d. If a back pressure check is used, the pressure drop due to the check must also be added to the total pressure drop for the valve.

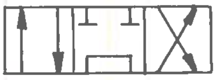
e. Correct for changes of viscosity.

EXAMPLE: (5-6)

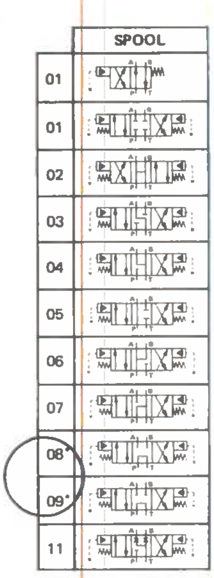
A particular D03 tandem center valve is passing 15 GPM using 100 SSU oil through all passing paths. What is the pressure drop through the paths in all conditions?

SOLUTION:

The configuration of the valve is found in the catalog. (Figure 5-56)

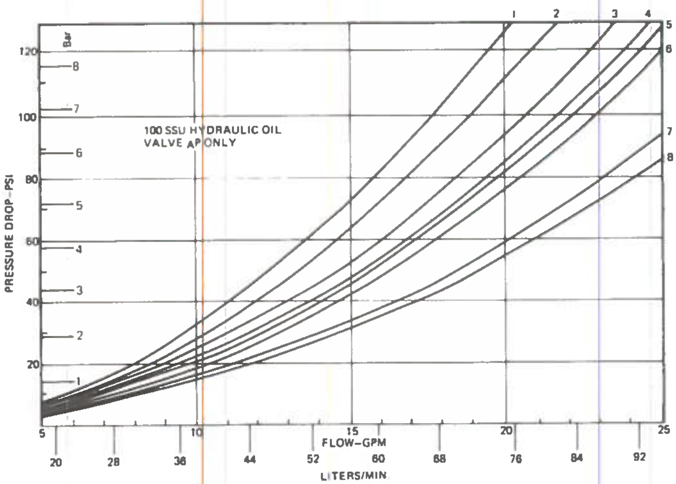


(Figure 5-56)



NOTE: 08 spool has all ports blocked when shifting valve spool to center position. 09 spool has all ports open when shifting valve spool to center position.

(Figure 5-57)



These curves were generated using 100 SSU hydraulic oil. For other viscosity the pressure drop will change per the following chart "B."

CHART "B" VISCOSITY CORRECTION FACTOR							
Viscosity (SSU)	75	150	200	250	300	350	400
Percentage of ΔP (Approx.)	93	111	119	126	132	137	141

(Figure 5-58)

The D03 directional control valve has a tandem center spool. From catalog data the spool configuration is found. (Figure 5-57)

Tandem spool configurations are either spool numbers 08 or 09. Because the pressure drop is the same in both spools, either may be used for pressure drop considerations. 08 was chosen here.

A chart of pressure drop vs. flow is now entered. (Figure 5-58)

The flow rate found along bottom horizontal, is entered at 15 gpm and a vertical line is drawn upward until it intersects one of the flow curves. (Figure 5-59)

For the 08 spool, chart A tells us what curves represent the pressure drop through particular flow paths. (Figure 5-60)

Since curve 2 and 1 are the only curves used, horizontal lines are drawn to the left from the intersection points. (Figure 5-61)

The pressure from the curve 1 intersection reads 73 psi. The pressure from the curve 2 intersection reads 65 psi.

EXAMPLE: (5-7)

A D10 solenoid controlled, pilot operated directional valve has an open center condition. Flow rate through the valve is 70 GPM. What is the pressure drop? (Figure 5-62)

SOLUTION:

Since this is an open center valve, it is questionable whether it would shift unless a back pressure check is incorporated. By inspecting the nameplate, we find a back pressure check was installed.

The spool number must be found. Through catalog data, it is determined to be number 02. (Figure 5-63)

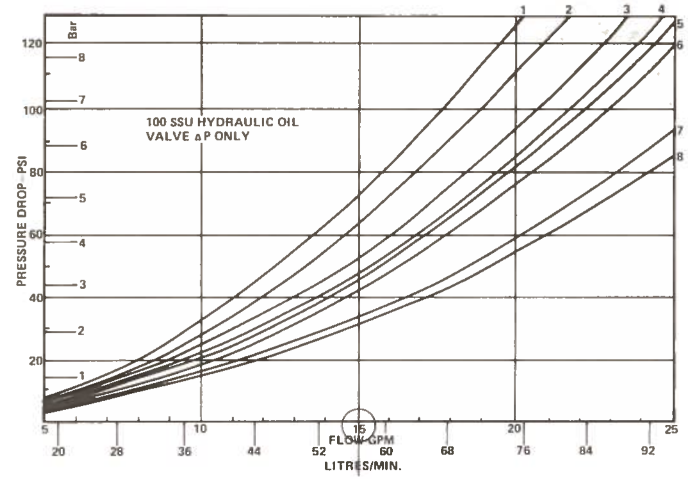
A chart of pressure drop is now entered. (Figure 5-64) The flow rate, along bottom horizontal is entered at 70 GPM and a vertical line is drawn upwards.

For the 02 spool, chart A reveals what curves represent the pressure drop through this valve with an open center spool. (Figure 5-65)

Curve 3 will give us the pressure drop P-A, P-B, and P-T. Curve 1 gives A-T, and curve 2 is B-T pressure drop. Drawing horizontal lines to the intersection points will determine the pressure drop. (Figure 5-66)

From the values found in the graphs, 65 psi is added to those flow paths that are influenced by the back pressure check. They will be ports with flow crossing the "P" port.

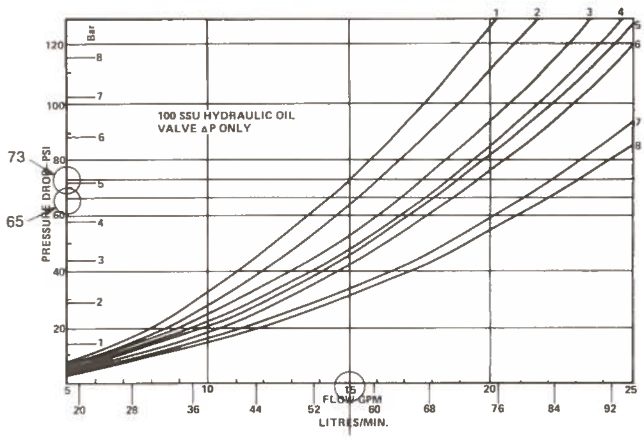
The solution is shown in (Figure 5-67).



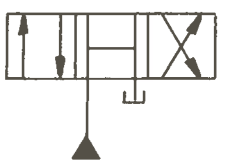
(Figure 5-59)

CHART "A" CURVE REFERENCE						
SPOOL CODE	MAX. FLOW	CURVE NUMBER				
		P-A	P-B	P-T	A-T	B-T
01	30	5	5		5	5
02	25	4	4	8	3	3
03	25	5	5		6	5
04	30	5	5		6	6
05	25	6	5		5	5
06	25	6	6		5	5
07	25	5	7	5	5	6
08	25	2	2	1	1	1
11	25	5	5		5	5
12	25	6	6		6	6

(Figure 5-60)



(Figure 5-61)

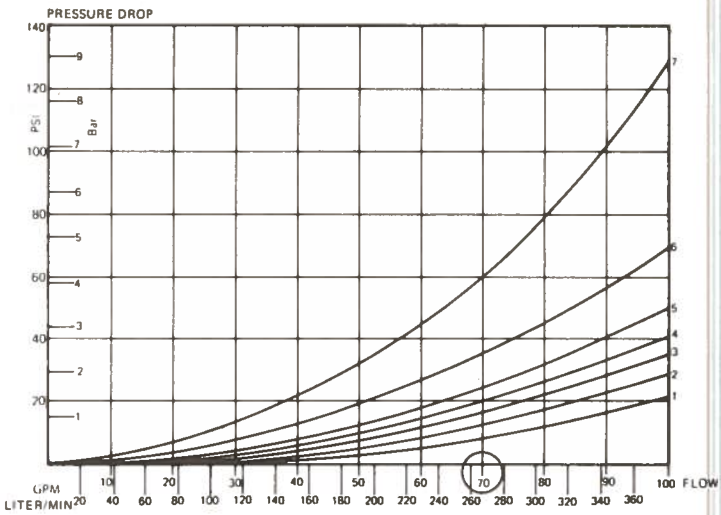


(Figure 5-62)

CODE	SYMBOL
1	
2	
3	
4	
5	
6	
7	
8 9	
11	
12	

(Figure 5-63)

If a viscosity was used other then that stated in the graph, a viscosity correction factor would be used.



(Figure 5-64)

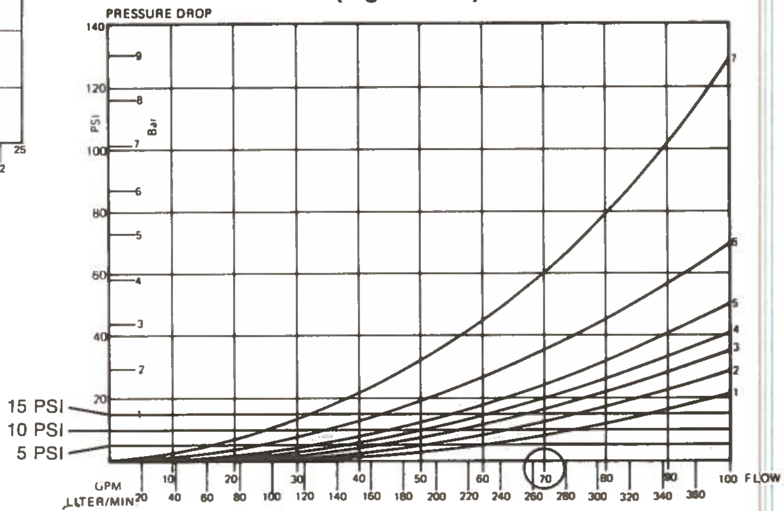
CHART A

SPOOL CODE	MAX. FLOW	CURVE NUMBER				
		P-A	P-B	P-T	A-T	B-T
01	110	4	4		2	3
02	110	3	3	3	1	2
03	110	4	4		1	3
04	110	4	4		1	2
05	110	3	4		2	3
06	110	3	3		2	3
07	100	4	3	7	2	2
08	100	5	5	6	2	3
09	100	5	5	6	2	3
11	110	4	4		2	3

These curves were generated using 100 SSU hydraulic oil. For other viscosity the pressure drop will change per the following chart "B".

CHART "B" VISCOSITY CORRECTION FACTOR							
Viscosity (SSU)	75	150	200	250	300	350	400
Percentage of ΔP (Approx.)	93	111	119	126	132	137	141

(Figure 5-65)



(Figure 5-66)

Pressure Drop PSI

	Flow	P-A	P-B	P-T	A-T	B-T
02	70 GPM	82	82	82	12.5	17

(Figure 5-67)

HEAT GENERATED

The heat generated in a valve is related to the pressure drop and the flow rate. The equation used for heat generation is:

Heat generation rate (HP) = GPM x PSI x .000583

Heat generation rate (J) =(LPM)x bar) / 248.8

Usually for any period of time two flow paths are passing flow. Because of this, both paths must be considered. For example, with a four way valve, when it is to one position form center, flow passes P-A, B-T. When shifted to the other position, P-B is passing as well as A to T.

EXAMPLE: (5-8)
Consider the D03 valve in example (5-6) and calculate the heat generation rate if the A solenoid is energized in 10 seconds. B for 5 seconds, none for 15 seconds.

SOLUTION:
The chart produced in the example was:

GPM	P-A	P-B	P-T	A-T	B-T
15	65	65	73	73	73

When the "A" solenoid is energized, P-A and B-T is passing. Heat generation rate:

Heat generation rate = GPM x PSI x .000583

= 15 GPM[(P-A) psi + (B-T)] x .000583

= 15 x (65 + 73) x .000583

= 1.20 HP for 10 seconds

When the "B" solenoid is energized, P-B and A-T is passing. The heat generation rate is:

Heat generation Rate = GPM x PSI x .000583

= 15 GPM (65 + 73) x .000583

= 1.20 Hp for 5 seconds

When no solenoids are energized, the passing flow path is P-T, therefore:

Heat = GPM x PSI x .000583

= 15 x 73 x .000583

= .64 HP for 15 seconds

The average heat generated is:

$$\text{Average Heat} = \frac{\sum_{n=1}^{\infty} Pn \times tn}{\sum_{n=1}^{\infty} tn}$$

$$= \frac{1.2 \text{ hp} \times 10 \text{ sec.} + 1.2 \text{ hp} \times 5 \text{ sec.} + .64 \text{ hp} \times 15 \text{ sec.}}{30 \text{ sec.}}$$

= .92 hp/cycle

CONCLUSION

In Chapter 5, check valve and directional valves were discussed. Both are considered to be distribution valves meaning that pressure drops should be 75 PSI or less.

Important formula's for check valves are:

Heat generation rate (P) = flow rate x total pressure drop x constant

$$Cv = Flow \sqrt{\frac{Specific\ Gravity}{\Delta P\ @\ Flow}}$$

$$Nr = 17{,}250\ \frac{Q\ (GPM)}{\sqrt{Cv \times \nu}}$$

$$Cv\ corrected = \frac{Cv}{Fv}$$

Pilot pressures required = Ratio x Blocked pressure

Important formula's for directional valves are:

Heat generation rate (HP)
= GPM x PSI Drop x 0.000583

Heat generation rate (J) = (LPM x bar) ÷ 248.8

For a directional valve, these considerations should be followed:

2-Position Valves

a. Determine the flow rates through the different paths in the valves.

b. With these flow rates, enter the graphs on the particular valve and extract the pressure drop.

c. Correct for change in viscosity.

3-Position Valves

a. Determine the type of center conditons on the particular valve in question

b. Determine the flow rate through the different passages in all conditions.

c. With the flow rates, enter the graphs of the particular valve and extract the pressure drop.

d. If a back pressure check is used, the pressure drop due to the check must also be added into the total pressure drop for the valve.

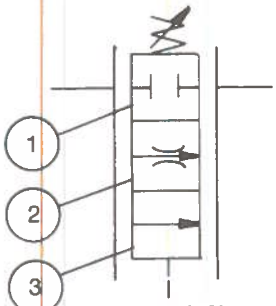
e. Correct for change in viscosity.

CHAPTER 6

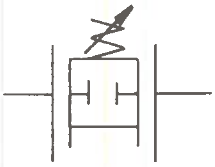
UNLOADING AND PRESSURE REDUCING VALVES



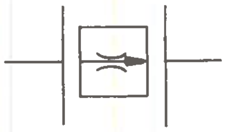
(Figure 6-1)



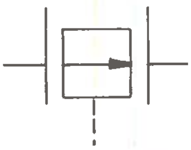
(Figure 6-2)



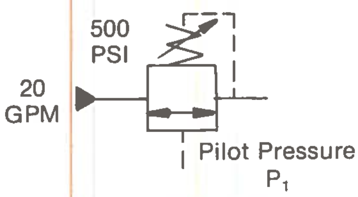
(Figure 6-3)



(Figure 6-4)



(Figure 6-5)



Time	P_1 (PSI)	Flow (yes or no)	Pressure Drop (if flow exists for 20 GPM)
1	100		
2	300		
3	700		

(Figure 6-6)

Time	P_1 (PSI)	Flow	Pressure Drop
1	100	no	X
2	300	no	X

(Figure 6-7)

In addition to the check and directional valves, valves such as unloading and pressure reducing can be said to direct fluid rather than cause large drops in energy. These valves will be covered in this chapter.

THE UNLOADING VALVE

An unloading valve is a normally closed or not passing pressure control valve which is used to unload the pump by directing its flow back to the tank at relatively low pressure. The symbol for this valve is shown in (Figure 6-1).

In order to more fully understand the working relationship of this valve with regards to pressure, let's rearrange the symbol to look like: (Figure 6-2).

This valve has three indiscrete areas to which the valve may be shifted. Area one shows a closed path. This section of the valve is in control when the pilot pressure is 100 psi (6.9 bar) less than the spring setting. (Figure 6-3)

When the pilot pressure is increased from about 100 psi (6.9 bar) less than or equal to 100 psi greater than the spring setting, the valve moves through the number (2) position. (Figure 6-4)

In this condition, the valve is metering flow and a large pressure drop may result. However, the valve, if applied correctly, should pass through this position quickly, therefore, it need not be considered.

When pilot pressure is increased 100 PSI (6.9 bar) above the approximate requirement of the spring, the valve spool moves to position (3). (Figure 6-5)

When in this position, the pressure drop is a function of the fluid, flow rate and design of the valve.

For the most part, the pressure drops through this type valve are not shown in catalog data. The distributor or factory must be consulted to get accurate values. An approximate figure of the pressure drop could be obtained by using the high vent pressure drop in a two stage relief valve of comparable size and design made by the same manufacturer.

The spring setting of this type valve should always be at least 100 psi (6.9 bar) less than the pilot pressure used to unload it. In this way, minimum pressure drop is ensured when the pump is unloaded.

EXAMPLE: (6-1)

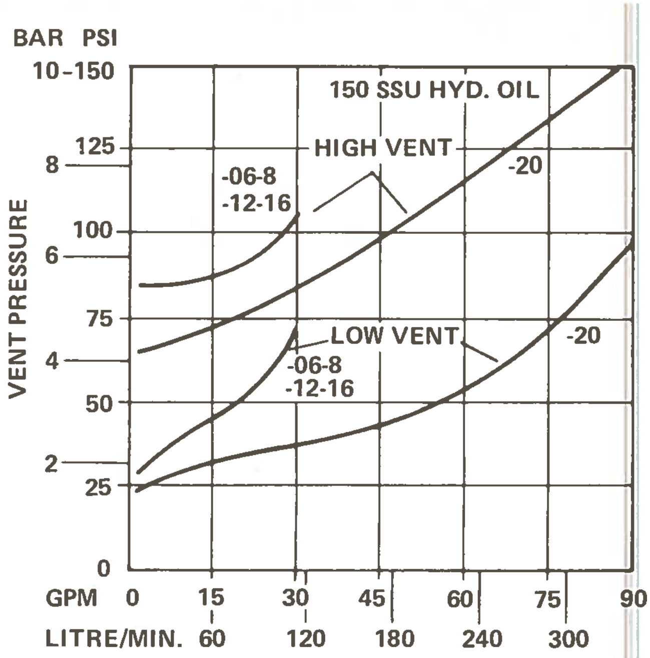
A P06 (¾ inch) unloading valve is employed in a circuit. What is the pressure drop at the various

times. (Figure 6-6)

SOLUTION:
For time (1) and (2), the pilot line pressure is less than the spring setting, therefore, the valve has no flow. (Figure 6-7)

For time (3) the pilot pressure is greater than 100 psi (6.9 bar) more than spring setting, therefore, the pressure drop data must be considered. Let's say the factory was not consulted and an approximate valve was needed. The pressure drop for the same manufacturer's two stage relief of the same size and design was obtained. The following graph shown the pressure drop. (Figure 6-8)

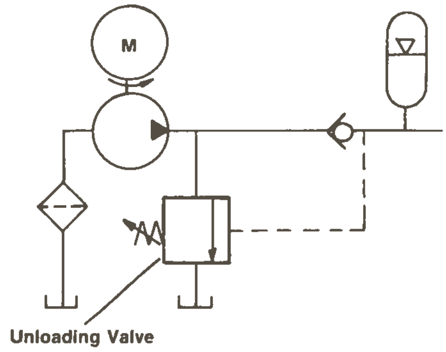
From the above graph, the pressure drop is shown to be about 90 psi (6.2 bar). This would be an approximate pressure drop for this ¾ inch unloading valve.



(Figure 6-8)

THE UNLOADING VALVE IN A CIRCUIT WITH AN ACCUMULATOR

In a typical circuit, when an accumulator is charged and work is not required from any part of the system, pump flow is unloaded to tank with the least possible pressure. (Figure 6-9)



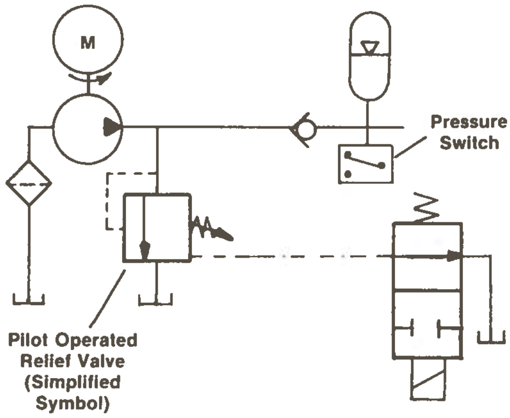
SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

(Figure 6-9)

In the circuit illustrated, an unloading valve allows flow to return to tank at low pressure once an accumulator is charged to the unloading valve setting. Frequently, unloading in this manner is only good for a few seconds at the most. With any leakage in the system downstream from the check valve, pressure in the accumulator will drop as fluid is dicharged to compensate. This results in the valve gradually closing. The opening to tank through the valve progressively becomes more and more restrictive until accumulator pressure drops below the valve cracking pressure. Up to this point, the pump has to develop more and more power as the valve closes.

Once the valve closes, the pump must therefore generate power to recharge the accumulator to the unloading valve setting. This means the electric motor applies power when it is supposed to be unloaded.

To keep a pump fully unloaded until it is required to re-charge an accumulator, an electric pressure switch can be used. (Figure 6-10)



SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

(Figure 6-10)

In the circuit illustrated a pressure switch senses accumulator pressure sending and cutting-out electrical signals at various pressure levels. The electrical signals are transmitted to a normally-open, solenoid operated 2-way valve which vents

and de-vents a pilot operated relief valve. When the accumulator is being charged, the pressure switch cuts out the signal venting the relief valve and unloading the pump. The setting of the pressure switch determines the pressure range within which a pump operates.

Using a pressure switch to vent a relief valve, results in a power unit being fully unloaded when system conditions dictate.

THE UNLOADING VALVE IN A HI-LO SYSTEM

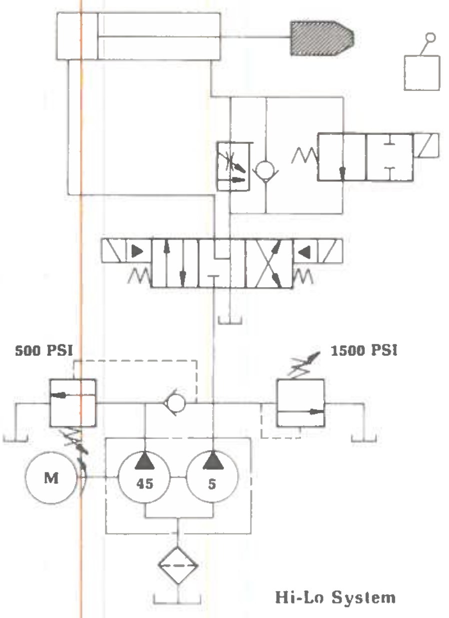
A hi-lo system consists of two pumps — one high volume, the other low volume. Both pump flows combine under low pressure to give a large rate of flow. Yet, when system pressure climbs above a certain value, the high volume pump is unloaded while the low volume pump does the work.

In the illustrated hi-lo system, the cylinder is not required to work through its entire stroke. Work is performed through the last few inches of stroke only.

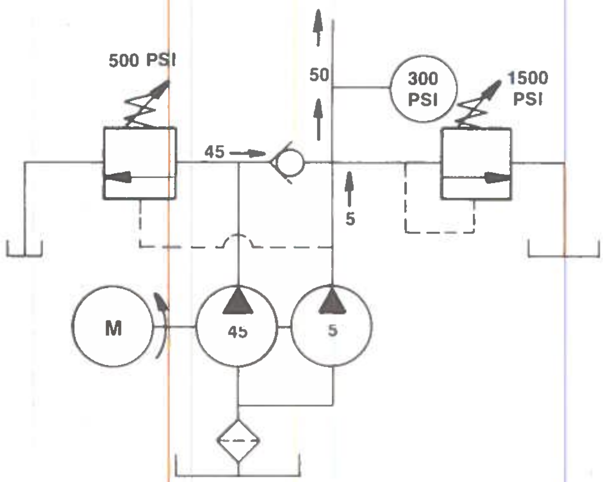
A flow rate of 4.5 GPM (17 LPM) flowing into the cylinder is required while the cylinder is doing its work. Yet, to reduce the amount of time getting to and from the work, 50 GPM (189 LPM) is desired. If a 50 GPM (189 LPM) pump were used in this system, 45 GPM (170 LPM) would dump back to tank over the relief valve while work is performed. This would be a gross mismatch of power generated to power used.

A hi-lo system satisfies the system demand by combining a 45 GPM (170 LPM) and 5 GPM (18.9 LPM) pump flow. (Figure 6-12) When the electric motor is turned on, the 45 GPM (170 LPM) passes through the check valve adding to the 5 GPM (18.2 LPM) 50 GPM (189 LPM) into the system extending the cylinder at a relatively low pressure. When the work load is contacted and work pressure is desired, pump pressure starts climbing toward the relief valve setting of 1500 PSI (103 Bar) (Figure 6-13) pressure level, the normally closed unloading valve opens allowing the 45 GPM (170 LPM) pump to unload while the 5 GPM (18.9 LPM) pump continues to work. This action eliminates unnecessary power generation by the 45 GPM (170 LPM) pump when it is not needed.

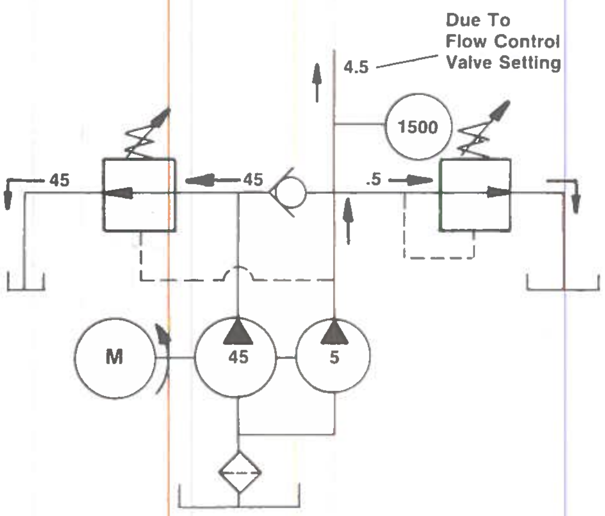
After work has been completed, the directional valve is shifted to retract the cylinder and pressure drops to a low level once again. This closes the unloading valve. The 45 GPM (170 LPM) adds to the 5 GPM (18.9 LPM) retracting the cylinder quickly.



(Figure 6-11)



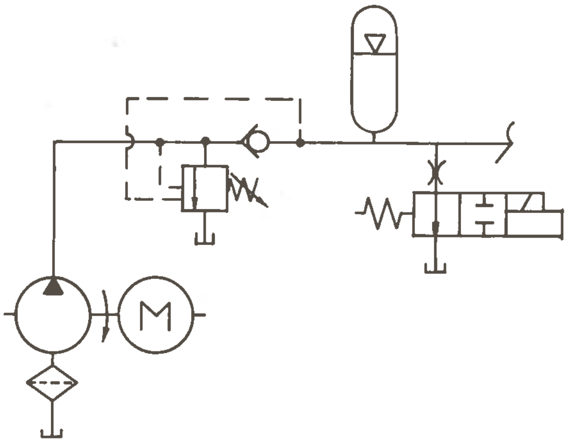
(Figure 6-12)



(Figure 6-13)

A hi-lo system gives high volume at low pressure and low volume at high pressure. In this way power generation is more evenly matched to actuator output. In other words, by applying the hi-lo system, the total energy input with fixed displacements pumps may be decreased in the following applications:

1. rapid advance and clamp (as in a jig or fixture)
2. rapid advance and press or compact (as in a press or compactor application)
3. rapid advance and slow retract of a cylinder



(Figure 6-14)

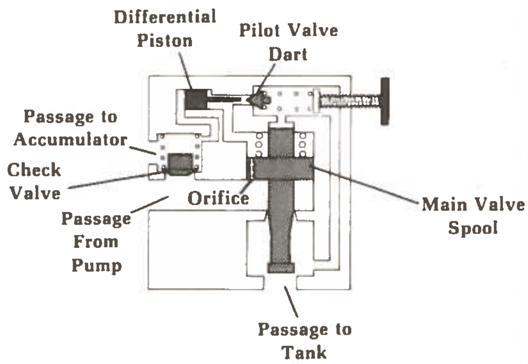
DIFFERENTIAL UNLOADING RELIEF VALVE

Instead of using a pressure switch and solenoid valve to vent a relief valve while an accumulator is charged, one hydraulic component can be used — a differential unloading relief valve. (Figure 6-14)

A differential unloading relief valve is specifically designed for use with accumulators. As its name implies, the valve unloads a pump over a differential pressure range.

WHAT A DIFFERENTIAL UNLOADING RELIEF VALVE CONSISTS OF

A differential unloading relief valve consists of a pilot operated relief valve, check valve, and differential piston in one valve body. The valve body includes pump, tank, and accumulator passages. (Figure 6-15)

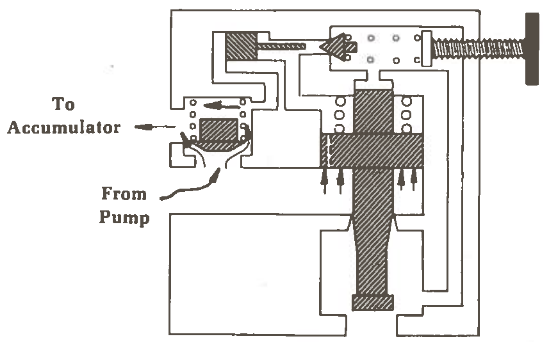


(Figure 6-15)

HOW A DIFFERENTIAL UNLOADING RELIEF VALVE WORKS

In a differential unloading relief valve, check valve and pilot operated relief valve operate in their usual manner. The accumulator charges through the check valve.

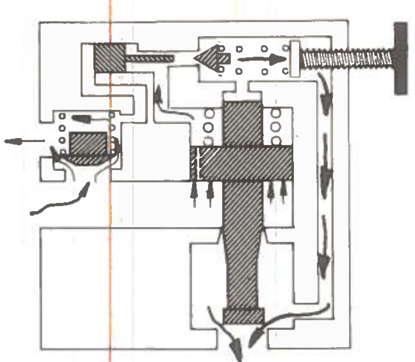
The differential piston is free to move within a bore opposite the pilot valve dart. Areas exposed to pressure at each end of the piston are equal. During the time the accumulator is being charged, pressure at each end of the piston is relatively equal. (Consider the pressure differential across the check valve to be negligible.) As a result, the piston does not move. (Figure 6-16)



Accumulator Being Charged

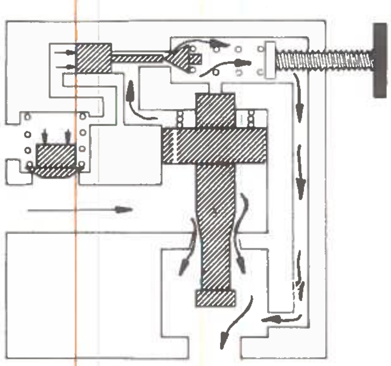
(Figure 6-16)

When a large enough pressure is present at the pilot valve dart, the dart is pushed off its seat. We have seen earlier that this action limits the pressure in the spring chamber of the main valve. With pressure limited in the spring chamber and also at one end of the differential piston, the piston is moved toward



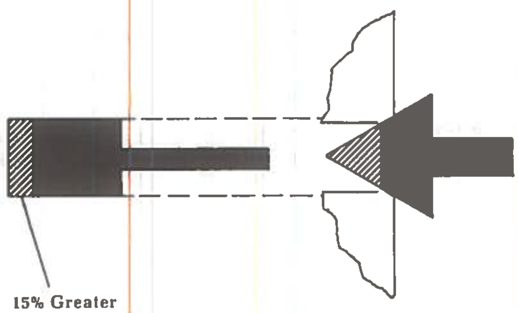
Pilot Dart Unseated

(Figure 6-17)



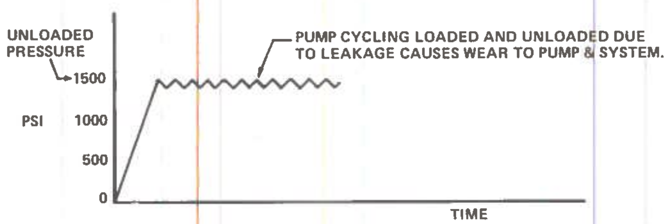
Relief Valve Vented
By Differential Piston

(Figure 6-18)



15% Greater
Area Exposed to
Pressure Than
Area of Dart

(Figure 6-19)



(Figure 6-20)

the pilot dart forcing the dart completely off its seat. This in effect releases the main spool spring chamber of pilot pressure, venting the relief valve and unloading the pump. At the same time the check valve closes so that accumulator flow cannot discharge through the relief valve. (Figure 6-17)

At this point, the accumulator's maximum pressure has been achieved and pump is unloaded. The differential piston is the key to unloading the pump until the accumulator discharges to a lower pressure. (Figure 6-18)

The differential piston has a 15% greater area exposed to pressure than the area of the pilot dart. Since force = pressure x area, the piston holds the pilot dart off its seat with a 15% greater force than the force which unseated the dart. This means that in order to reseat the pilot dart, the spring must acquire a 15% greater force from somewhere or, it must wait until the pressure in the system falls off 15%. Of course, the dart reseats when system pressure falls off 15%. (Figure 6-19)

In this way, a differential unloading relief valve allows a pump to unload when an accumulator is charged and to remain unlaoded until re-charging is required.

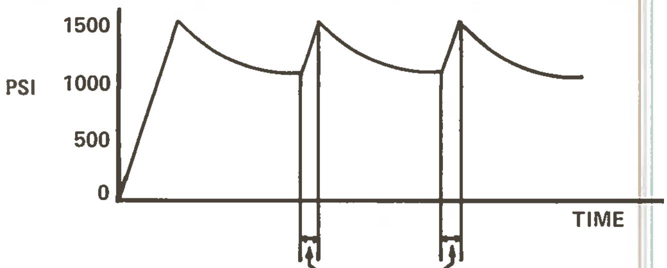
A limitation of a differential unloading relief valve is that the valve's secondary pressure is fixed because the difference in areas between piston and pilot dart is fixed. This is frequently 15% and in some cases 30% of the pilot valve setting. For example, a differential unloading relief valve with a 15% differential will unload between 1000 PSI (69 bar) and 850 PSI. (59 bar) with its pilot valve adjusted for 1000 PSI (69 bar). Or with its pilot adjusted for 2000 PSI (138 bar), it will unload between 2000 PSI (138 bar) and 1700 PSI (117 bar).

WHY A DIFFERENTIAL UNLOADING VALVE IS NEEDED

In a typical circuit with an unloading valve, the pump charges the accumulator to a set pressure and then unloads. If there were no leakage in the system, the pump would stay unloaded. However, if leakage exists, oil will flow from the accumulator into the system. This means a drop in pressure. Consequently, as soon as the pressure drops, the pump loads. If the leakage is substantial, the pump may never unload. A typical graph of pressure verses time is shown. (Figure 6-20)

With a differential unloading relief, the pump will unload, just as it had before. Consider system leakage to be the same. The accumulator discharges

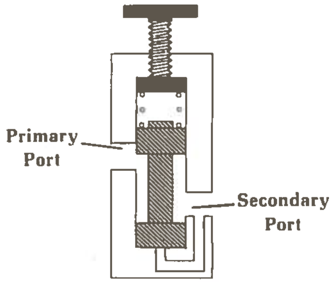
a small amount of flow to make up leakage. This causes the pressure to drop. However, due to the valve designed, the pump will not load again until the pressure decreases 15 to 30% below the unload point (the 15 or 30% is set by manufacturer). This saves wear and tear on the pump as well as the system. The graph of pressure verses time for the system becomes (Figure 6-21).



(Figure 6-21)

PRESSURE REDUCING VALVES

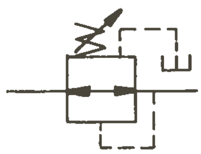
Pressure reducing valves are the only normally open pressure control valve in a typical hydraulic circuit. They sense downstream pressure and regulate it to a predetermined value. This pressure reduction is done by providing a variable restriction to flow. The energy caused by the resulting pressure drop is sensed as heat. If the pressure downstream of the valve drops off, the restriction will open and allow flow and pressure to build once again. (Figure 6-22)



Pressure Reducing Valve
(Figure 6-22)

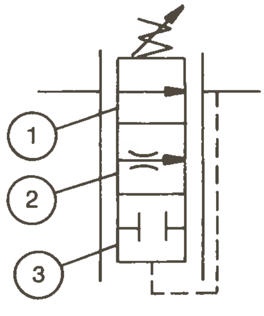
These valves are used to maintain reduced pressures in certain branches of the circuit. When the main supply pressure is below the equivalent spring pressure, the valve will remain open with the flow relatively unrestricted. Only enough flow is passed to the outlet to maintain the downstream pressure. It should be noted that if the restriction closed completely, leakage internal to the valve may cause pressure to build up downstream. To prevent this, a continuous bleed to tank through the external drain ensures that the variable restriction is always somewhat open.

The drain flow may vary anywhere from ⅓ GPM, (1.25 LPM) in small valves, to 1.5 GPM (5.67 LPM) in large valves. Because of this, drain lines must be sized to handle flows of this size. If drain lines are excessively small, pressure may build above the spool preventing its' closure. This means pressure will NOT be reduced.

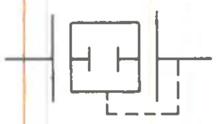

(Figure 6-23)

The symbol for the pressure reducing valve is shown (Figure 6-23). The spring setting determines the maximum pressure that will exist downstream, no matter what pressure exists at its' upstream port. The valve may be drawn functionally as in (Figure 6-24).

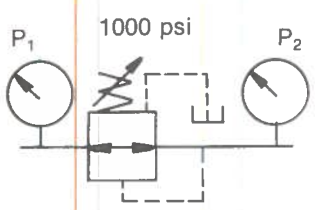
(Figure 6-24) shows the valve in a condition when the upstream pressure equals the downstream pressure, both of which are below the equivalent spring setting of the valve. Pressure drop through the valve is just a function of fluid, its velocity and the valve. In this condition, (#1), the valve has little pressure drop. This is the best condition for the valve to be in when the flow is occuring.



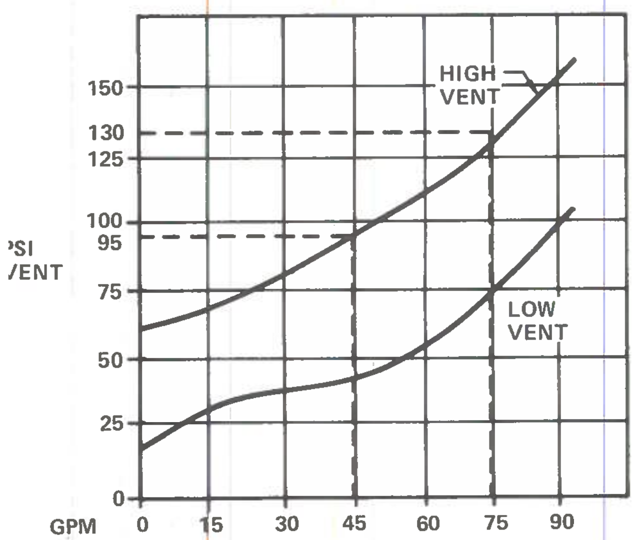
(Figure 6-24)



(Figure 6-25)



(Figure 6-26)



(Figure 6-27)

The center position will exist for only a short time. Therefore, it need not be considered.

The third position (Figure 6-25) is the condition at which the valve is reducing pressure.

The condition takes over when the pressure at the downstream side of the valve equals the spring setting. Constant pressure is maintained downstream with the resulting pressure drop sensed as heat.

Once again, the pressure drop through the valve at position one may not be conveniently found in catalog data. The distributor or factory must be consulted. If data is not easily obtained, the procedure outlines the section on unloading valves should be applied.

EXAMPLE: (6-2)

A P10 (1¼) pressure reducing valve is employed in a circuit. What is the pressure drop at various times (Figure 6-26).

FIND:

Time	P_1 (PSI) (bar)	P_2 (PSI) (bar)	Flow Rate (GPM) (LPM)
1	?	500/34.5	75/283.8
2	?	800/55.2	45/170.3
3	1300/89.6	?	0

SOLUTION:

The valve will be in position (1) in both time 1 and 2. This means that P_1 is a function of the fluid, the flow rate and the design of the valve. Catalog data cannot be found. An approximate answer is needed quickly so the comparable 2 stage relief characteristics are consulted. The high vent is used to find the pressure drop. The curve is shown in (Figure 6-27).

For time 1, the curve is entered at the 75 GPM (283.8 LPM) line and the pressure found to be equal to 130 psi (9 bar). The pressure drop at time 2 is entered at 45 GPM (170 LPM) and found to be 95 psi (6.7 bar).

The pressure P_1 @ time 1 is equal to:

P_1 @ 1 = Pressure drop + P2

P_1 @ 1 = 130 + 500 = 630 psi (43.4 bar)

The pressure of P_1 @ time 2 is:

P_1 @ 2 = Pressure drop + P_2

P_1 @ 2 = 95 + 800 = 895 psi (61.7 bar)

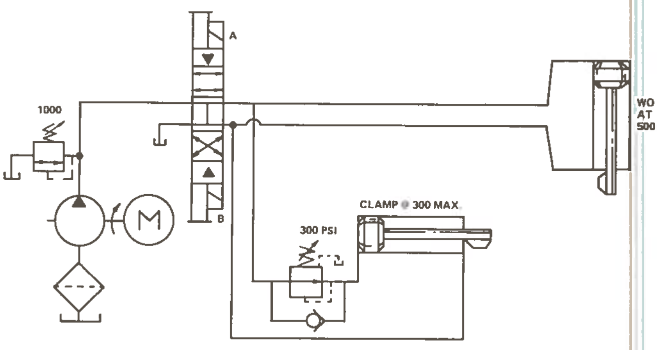
At time 3 is the only time the valve is in the 3 position because the upstream pressure has exceeded the spring setting of the valve. This means that P_2 takes on the setting of the valve meaning.

P_2 = 1000 psi (69. bar)

The excess pressure is sensed as heat.

Time	P_1 (PSI) (bar)	P_2 (PSI) (bar)	Flow Rate (GPM (LPM)
1	630/43.4	500/34.5	75/283.8
2	895/61.7	800/55.2	45/170.3
3	1300/89.6	1000/69	0

A common use of the pressure reducing valve is to limit pressure in one cylinder is operating at a higher pressure. Consider the following circuit. (Figure 6-28)



(Figure 6-28)

In this circuit, the clamp will extend at 100 psi (6.9 bar) and must clamp at a maximum pressure of 300 psi (20.7 bar). The work cylinder pressure is 500 psi (34.5 bar) ± 5%. This means that the pressure reducing valve must be placed ahead of the clamp cylinder as shown. The valve is in position (1) while the clamp cylinder is extending (low pressure drop) (1) and is in position (3) when the clamp has reached the part (reducing 500 psi to 300 psi). When in position (3) excess pressure (500 psi directed to tank is senses as heat). Without the reducing valve, the pressure in the clamp would climb to the pressure in the work cylinder.

It is good engineering practice to use this valve in such a way that during the flowing condition of the valve (condition 1) the downstream pressure would be at least 100 psi (6.9 bar) less than the spring setting of the valve. This keeps the internal moving parts in the extreme condition so that pressure drop through the valve is kept at a minimum.

CONCLUSION

In Chapter 6, unloading, differential unloading and pressure reducing valves were considered.

It is recommended that an unloading valve be used with a double pump to unload a high volume pump to tank at low pressure. It's pressure drop at various times was discussed.

With a differential unloading valve, an accumulator may be charged and then the pump unloaded. A predetermined deadhead is present (15 or 30%) which is inherent to the design of the valve.

The pressure reducing valve is usually the only normally open pressure control valve in the system. It is used to reduce the pressure in a particular leg of a circuit.

For all the above valves, leakage rates, and pressure drops should be obtained and applied when designing a system.

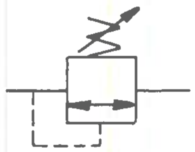
CHAPTER 7

RELIEFS, COUNTERBALANCES & BRAKE VALVES

When employing pressure control valves, thoughtful circuit design should be exercised as overall efficiency of the system could be greatly reduced.

Valves that fit into this category are sequence, counterbalance, flow control, relief valves and pressure reducing.

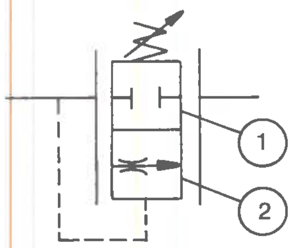
One valve body depending on where it is placed in the system, takes on different names. For instance, a normally closed pressure control valve can be a relief or a counterbalance valve. Each will be discussed in this chapter. Let us start the discussion with the relief valve.



(Figure 7-1)

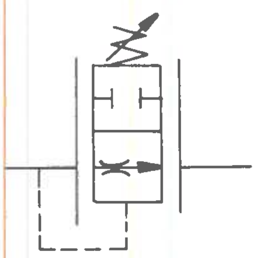
RELIEF VALVE

This type of pressure control valve is used to provide a maximum pressure limit at the pump. It is a normally closed pressure control valve, held closed by a spring until the pilot pressure exceeds this spring setting. The symbol for this valve is shown in (Figure 7-1).



(Figure 7-2)

A more explanatory diagram of a direct operated relief valve could be drawn as shown in (Figure 7-2).



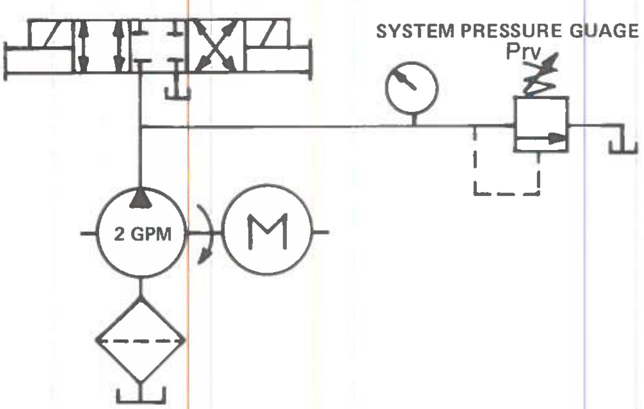
(Figure 7-3)

This valve will stay in position (1) until the system pressure approaches the relief valve setting. The value at which the valve will begin to pass flow is a function of the body and internals. Consider the valve is relieving 56.7 LPM (15 GPM) at 100 bar (1450 psi). In order that the valve may perform this function, the spool must begin opening below the 100 bar (1450 psi) mark. Once the valve begins to open, a small portion of the flow is directed back to tank. As the pressure increases, the valve opens further and less flow enters the system and more flows through the valve. At the set point pressure, most of the flow will be directed through the valve. If, for some reason, flow was increased, pressure will rise above or "over-ride" the 100 bar (1450 psi) mark. This type of operation exists due to the compression of the set point spring. Just as force increases when you compress a spring, pressure must increase as you compress the spring to move to spool in such a manner as more flow can pass through the valve.

The typical characteristics of a particular valve design are important and can be obtained from catalog data. Below is such a curve for a direct operated relief valve. Let us use the given data to work an example problem.

EXAMPLE: (7-1)

A particular relief valve is set to relieve a 2.0 GPM(7.6 LPM) pump at 2000 psi (140 bar). The circuit configuration is shown in (Figure 7-4).



(Figure 7-4)

The relief was set when the directional valve was centered. Using the catalog data provided, what is:

a. maximum system pressure with negligible flow passing through the relief valve.

b. system flow at 1740 psi (120 bar)

c. the system pressure, if 3.0 GPM (11.4 LPM) was relieved through the valve.

SOLUTION:

The operating curve of the valve must first be found. This is found by intersecting the 2 GPM (7.6 LPM) and 2000 psi (140 bar) lines. Where the "X" is placed depicts the operating curve.

a. Negligible flow will pass through the valve when the operating curve passes through 0. The pressure is read directly at the intersection and found to be 1500 psi (106 bar)

b. The relief flow must first be found. The intersection of the 1740 psi (120 bar) and the operating line corresponds to a 1.0 GPM (3.78 LPM) flow. From this, we may write:
   system flow = pump flow - relief flow
   system flow = 2 GPM - 1 GPM
   = 1 GPM (3.78 LPM)

c. System pressure at 3.0 GPM (11.4 LPM) would be 2400 psi (165.5 bar).

The heat generation rate in this system can be calculated by employing the power equation.

Heat generation rate (HṖ) = flow over relief (GPM) x pressure (psi) x .000583

In the preceeding example with the entire pump flow going through the relief valve, the heat generation rate is:

Heat rate (HP) = 2 GPM x 2000 psi x .000583

Heat generation rate = 2.332 HP (1.74 KW)

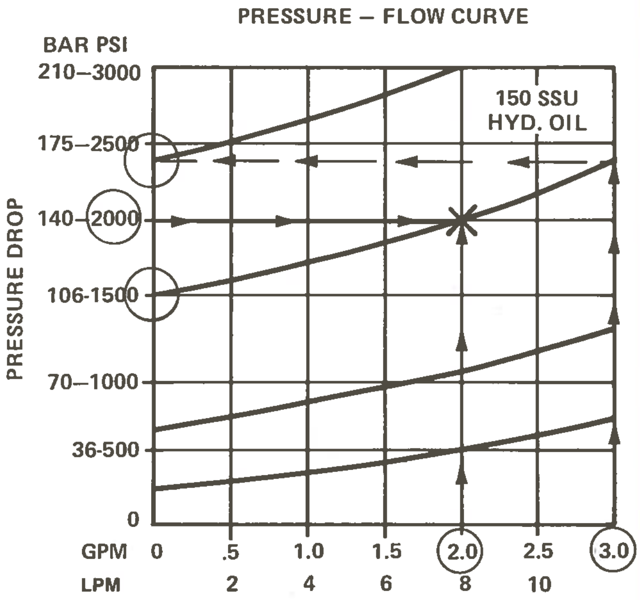
If it is desired to have a multitude of relief valve settings, or if a better performing relief valve is needed, a pilot operated relief valve may be used.

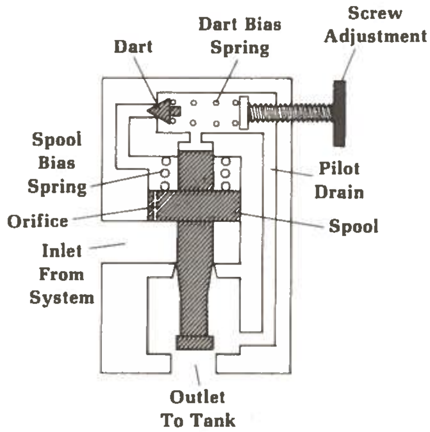
PILOT OPERATED RELIEF VALVE

A pilot operated relief valve basically consists of a main valve and a pilot valve. The valve operates by using both spring and fluid pressure to bias a main spool. The spring pressure is relatively light. (Figure 7-5)

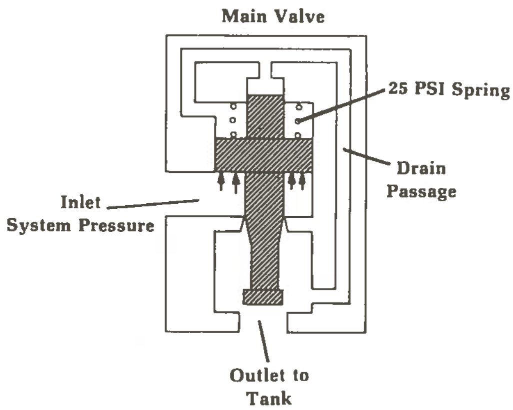
HOW A PILOT OPERATED RELIEF VALVE WORKS

To understand the operation of a pilot operated





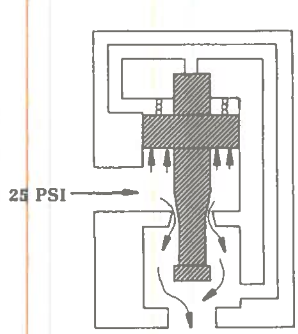
(Figure 7-5)



(Figure 7-6)

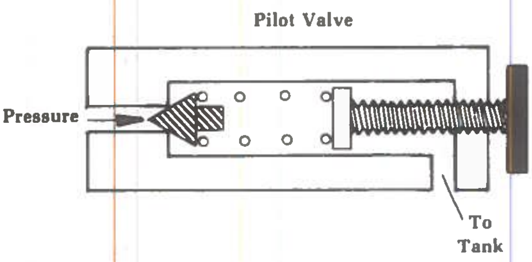
relief valve, we will look at the independent operation of the main valve and the pilot valve. The main valve spool is biased by a relatively light spring. The stem of the main valve spool plugs the outlet to the tank. System pressure acts on the area under the spool skirt. Any leakage passed the spool is internally drained to tank through a passage in the valve body. (Figure 7-6)

If the spring biasing the main valve spool has a value of 25 PSI, (1.7 bar) the spool will be pushed up and system flow will pass to tank when system pressure reaches 25 PSI (1.7 bar). In this way, the valve functions as a simple, spring-biased pressure control valve. (Figure 7-7)



(Figure 7-7)

The movable part of the pilot valve is a poppet which looks like a dart. The area of the dart exposed to hydraulic pressure is relatively small. The spring which biases the dart on its seat is rather stiff. The combination of small area and stiff spring means that the dart will remain seated until a high pressure is reached. (Figure 7-8)



(Figure 7-8)

If the spring biasing the dart has a value of 975 PSI (67 bar) the dart will remain seated until this pressure is reached. At this time, the dart will unseat and flow will pass to tank. Pressure is limited to approximately 975 PSI (67 bar). In this manner, the pilot valve also acts as a simple, spring-biased pressure control valve. (Figure 7-9)

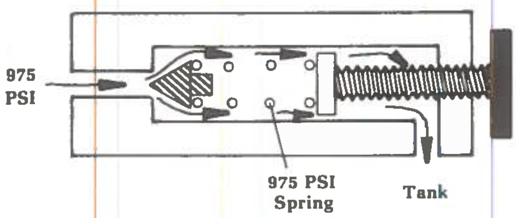


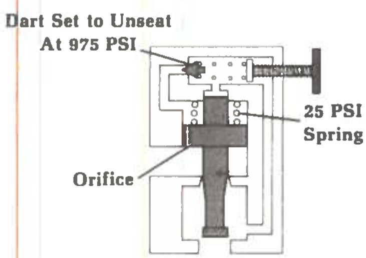
Figure 7-9)

The pilot valve is a simple, normally closed pressure control which handles small flows at high pressure. The main valve is also a simple, normally closed pressure control which handles large flows at a low pressure. By combining the two, large flows can be handled at high pressure.

In a pilot operated relief valve, the main valve spool is biased by light spring pressure and fluid (pilot) pressure in the spring chamber. The maximum fluid pressure which is allowed to bias the spool is determined by the setting of the pilot valve.

To allow pressure to accumulate in the spring chamber, an orifice or hole is drilled through the main valve spool skirt.

To illustrate the operation of a pilot operated relief valve, assume that the spring biasing the main valve spool has a value of 25 PSI (1.7 bar) and that the pilot valve will limit pilot pressure in the spring chamber to 975 PSI (67 bar) (Figure 7-10)

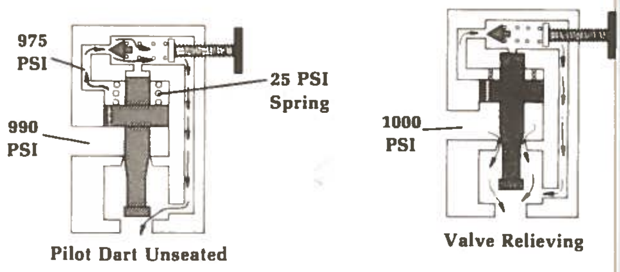


With a system pressure of 800 PSI, 800 PSI (55 bar) is acting to push the spool up. 800 PSI is transmitted through the orifice to the spring chamber acting to hold the spool down. The areas exposed to pressure

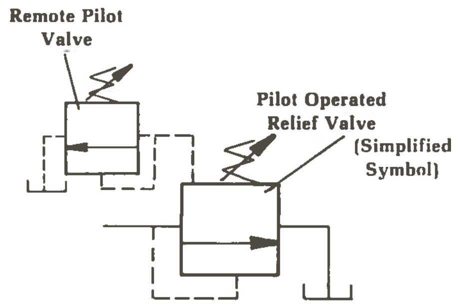
on either side of the spool skirt are equal. Therefore, the spool is balanced except for the 25 PSI (1.7 bar) spring. Consequently, there is a hydraulic pressure of 800 PSI (55bar) trying to unseat the spool and a total hydraulic and mechanical pressure of 825 PSI (57 bar) keeping the spool seated.

With a system pressure of 990 PSI, 990 PSI (68 bar) at the bottom of the spool acts to push the spool up. Since the pilot valve is set to limit the fluid pressure in the spring chamber to 975 PSI, (67 bar) the pilot valve dart is unseated and pilot pressure above the spool skirt is 975 PSI (67 bar). This is a total hydraulic and mechanical pressure of 1000 PSI (69 bar) acting to hold the spool down; the total pressure acting down is still more than the pressure acting up. The maximum pressure which can bias the spool in the down position is 1000 PSI (69 bar). If the pressure below the spool skirt attempts to rise above 1000 PSI, (69 bar) the spool will crack open and flow will pass to tank.

In our example, up to a pressure of 975 PSI (67 bar) total mechanical and fluid pressure biasing the spool will be 25 PSI (1.9 bar) more than system pressure. Between 975 PSI (67 bar) and 1000 PSI (69 bar), the difference becomes less until at any pressure over 1000 PSI (69 bar) main valve spool is unseated.



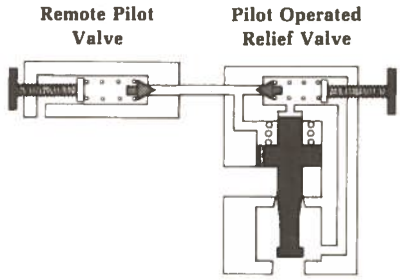
(Figure 7-10)



(Figure 7-11)

REMOTE PILOT ADJUSTMENT

Since fluid pressure is used to bias the main valve spool, a pilot operated pressure control valve can be adapted for remote adjustment. In the illustration, a remote pilot valve is used in conjunction with a pilot operated relief valve. With the remote pilot valve set for 600 PSI (41 bar) the main pilot valve set for 1000 PSI (69 bar) and the main pilot valve set for 1000 (69 bar), the main pilot valve determines relief setting. (Figure 7-12)



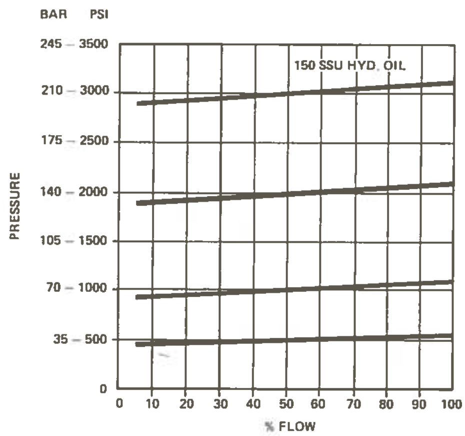
(Figure 7-12)

With a remote pilot adjustment, a pilot operated relief valve can be adjusted downward from the main relief setting. A remote adjustment above the mainsetting will not affect the valve operation.

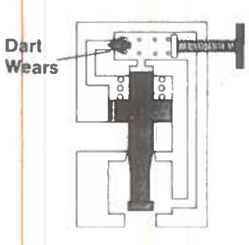
CRACKING PRESSURE PILOT OPERATED RELIEF VALVE

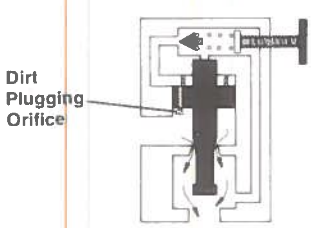
Since the main spring of a pilot operated relief valve is relatively light, the valve cracking pressure is closer to full flow pressure than in a simple relief valve (Figure 7-13)

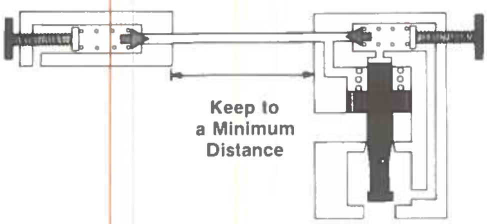
The illustrated performance curve for a pilot operated relief valve points out that the valve cracks open and reaches full flow within 100 psi (6.9 bar).



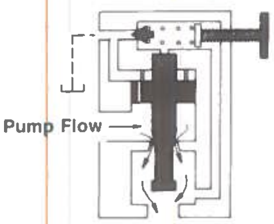
(Figure 7-13)



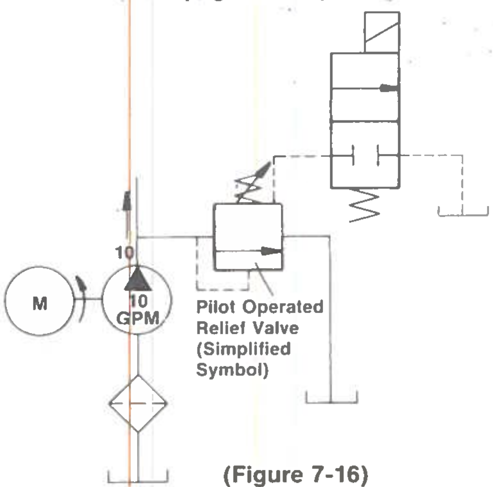




(Figure 7-14)



(Figure 7-15)



(Figure 7-16)

Whereas, with the simple relief valve seen earlier, the valve cracked open and reached full flow within 200 PSI (13.8 bar) while handling the same flow.

OPERATING PROBLEMS WITH A PILOT OPERATED RELIEF VALVE

Certain problems can arise while using a pilot operated relief valve. Many problems can be traced to dart wear, orifice plugging, and excessive line lengths between remote and main pilot valves.

As a pilot operated relief valve operates, the dart in the pilot valve can wear as it moves on and off its seat. This causes leakage from the main spring chamber back to tank. If wear, and consequently leakage become excessive, sufficient pilot pressure above the spool skirt will not be maintained. As a result, valve and system will become erratic.

In cases where systems are not properly protected with filters, dirt can plug the orifice in the spool. This means pressure can no longer pass the pilot-spring chamber on top of the spool. When this occurs, the valve will actuate one time. The spool will not close or will close very slowly since fluid pressure has a difficult time transmitting through the orifice.

In some instances, a problem occurs where excessive line lengths exist between main valve and a remote pilot adjustment. It appears that the farther a remote pilot valve is removed from its main valve the more chance of pressure pulsations, fluid compressibility, and valve springs interacting to generate irritable noises and excessive vibration. For this reason, it is usually recommended that line lengths be kept to a minimum.

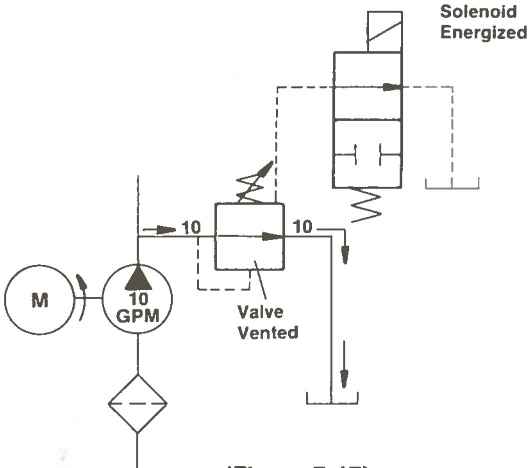
UNLOADING THROUGH A PILOT OPERATED RELIEF VALVE

A common way of unloading pump flow during idle time is "venting". (Figure 7-15)

Venting a relief valve refers to releasing the fluid pressure biasing the main spool of a pilot operated relief valve. By releasing this pilot pressure, the only pressure holding the spool closed is the relatively light pressure of the spring. This results in the pump applying a relatively low pressure to return its flow to tank. (Figure 7-16)

In the circuit illustrated, a solenoid operated directional valve is connected to the vent line of a pilot operated relief valve. With the solenoid de-energized, maximum pump pressure is determined by the relief valve setting. When the solenoid is

energized during idle time, pump flow returns to tank at whatever pressure it takes to overcome the relatively light spring biasing the spool. (Figure 7-17)



(Figure 7-17)

HIGH AND LOW VENT

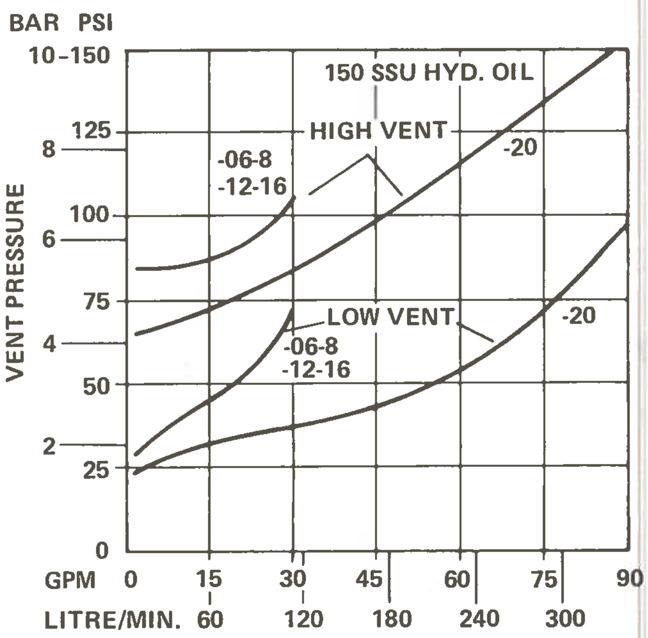
Pilot operated relief valves can be equipped with either a high or low vent spring.

A low vent spring exerts minimal pressure on the main spool of a pilot operated relief valve. This allows pump flow to return to tank through the valve with the least possible resistance during venting. A low vent spring is the standard main spring of a pilot operated relief valve.

A high vent spring is a stiffer spring biasing the main spool. It is a greater resistance to tank during venting.

If venting is done to allow pump flow to return to tank at a low pressure, the use of a stiff spring may be questioned. Even though a high vent spring is a greater resistance to tank, it is found in systems where better valve response time during deventing is required. It also ensures that the valve will close especially if any back pressure is present in the tank line.

A comparison of high and low vent spring operation is given in the illustrated graph. (Figure 7-18)



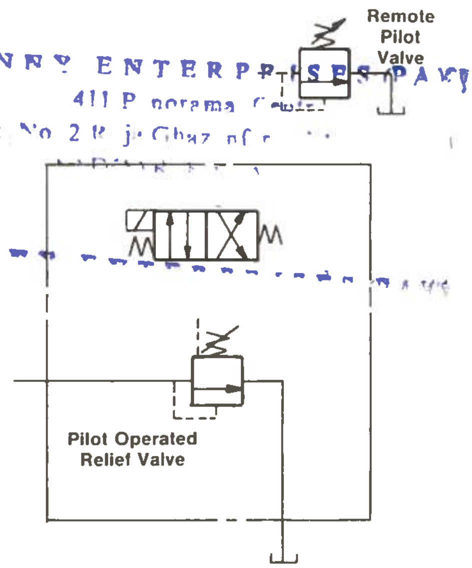
(Figure 7-18)

SOLENOID OPERATED RELIEF VALVE

In the description of a pilot operated relief valve, it was indicated that the pilot valve setting determined at which point the valve would limit pump pressure. It was also shown that a remote pilot valve connected to the main valve spring chamber would control relief valve operation as long as its setting were lower than the main pilot valve. A solenoid operated relief valve takes advantage of this arrangement.

WHAT A SOLENOID OPERATED RELIEF VALVE CONSISTS OF

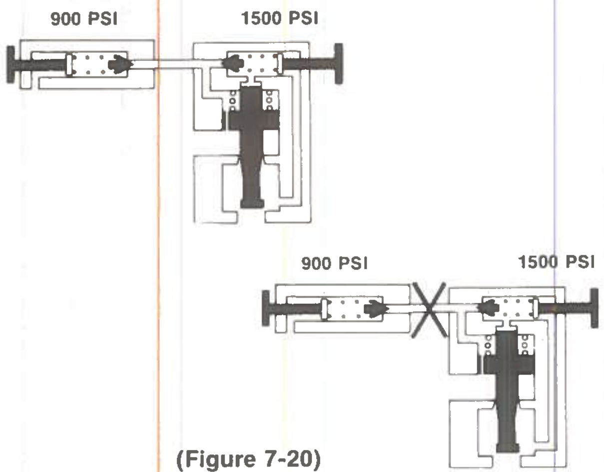
A solenoid operated relief valve consists of a pilot operated relief valve, directional valve, and remote pilot valve. The directional valve is mounted on top of the relief valve body. The remote pilot valve can be mounted on a fixture or panel near the main valve. (Figure 7-19)



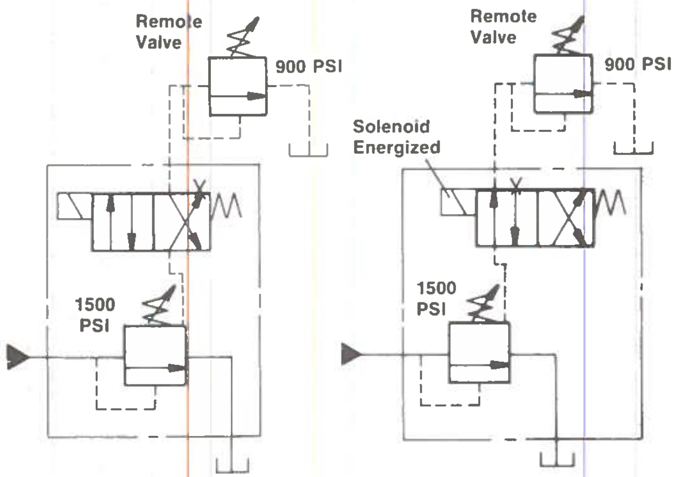
(Figure 7-19)

HOW A SOLENOID OPERATED RELIEF VALVE WORKS

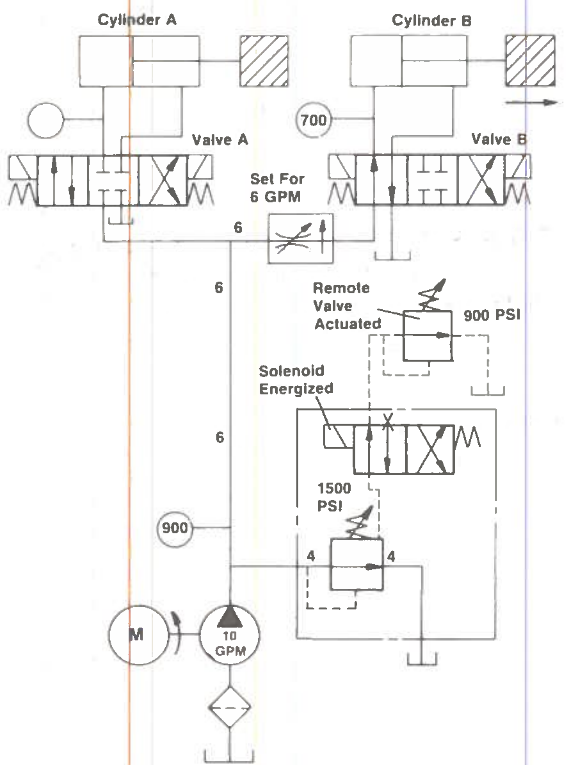
A solenoid operated relief valve changes setting as



(Figure 7-20)



(Figure 7-21)



(Figure 7-22)

the directional valve mounted on top of its housing is shifted.

In the illustration, a remote pilot valve is connected to a pilot operated relief valve. With the remote valve set for 900 PSI (62 bar) and the main pilot valve adjusted to 1500 PSI (103 bar), the remote pilot valve controls the relief valve operations. However, if the remote valve were disconnected, the main pilot valve would determine relief pressure. The action of connecting and disconnecting the remote pilot valve in a solenoid operated relief valve is performed by the directional valve.

With the spring offset directional valve de-energized, remote pilot valve is disconnected from the main valve. Maximum pump pressure is determined by the setting of the main pilot valve which in this case is 1500 PSI (103 bar). With the directional valve shifted, remote and main valves are connected. Maximum pump pressure is now limited to 900 PSI (62 bar).

SOLENOID OPERATED RELIEF VALVE IN A CIRCUIT

With a solenoid operated relief valve in our two-cylinder circuit, power generation can be more evenly matched when cylinder B is working. (Figure 7-22)

The main pilot valve is set to limit pump pressure to 1500 PSI (103 bar); the remote pilot valve is set for 900 PSI (62 bar).

When cylinder A is working, electric circuitry controlling the solenoid operated relief valve is arranged so that its directional valve is de-energized. Maximum pump pressure is limited to 1500 PSI (103 bar), Pump discharges at 10 GPM (37.8 LPM) at 1400 PSI (97 bar). Cylinder A uses 7.6 HP (5.6 KW) and pump accepts 8.2 HP (6.1 KW).

When cylinder B is working, the directional valve of the solenoid operated relief valve is energized. This action causes the remote pilot valve and main valve to connect limiting pump pressure to 900 PSI (62 bar). At this point, 5.3 hydraulic horsepower (3.9 KW) is being generated and 4.1 HP (3 KW) is used by cylinder B. This is a closer match than if the relief valve setting were kept at 1500 PSI (103 bar).

In the circuit just described, the solenoid actuated, spring offset valve connected and disconnected a remote pilot valve to the main spring chamber of a pilot operated relief valve. If the remote pilot port of the valve were connected to tank instead of to a remote pilot valve, the relief valve would be vented

when the directional valve was shifted. This is commonly done.

Solenoid operated relief valves are frequently equipped with a 3-position directional valve. This allows three different relief valve settings — one for each directional valve position.

Referring to our two cylinder circuit once again, the illustration shows that the solenoid operated relief valve now has a directional valve with three positions. The directional valve has a tandem center. (Figure 7-23)

Electrical circuitry is arranged so that when cylinder A is working, the directional valve is not connecting the main relief spring chamber with anything. Maximum pump pressure is therefore limited to 1500 PSI (103 bar).

When cylinder B works, the relief directional valve joins remote pilot valve to spring chamber limiting pump to 900 PSI (62 bar).

When work is completed, the directional valve is centered connecting main spring chamber to tank. The relief valve vents, unloading the pump.

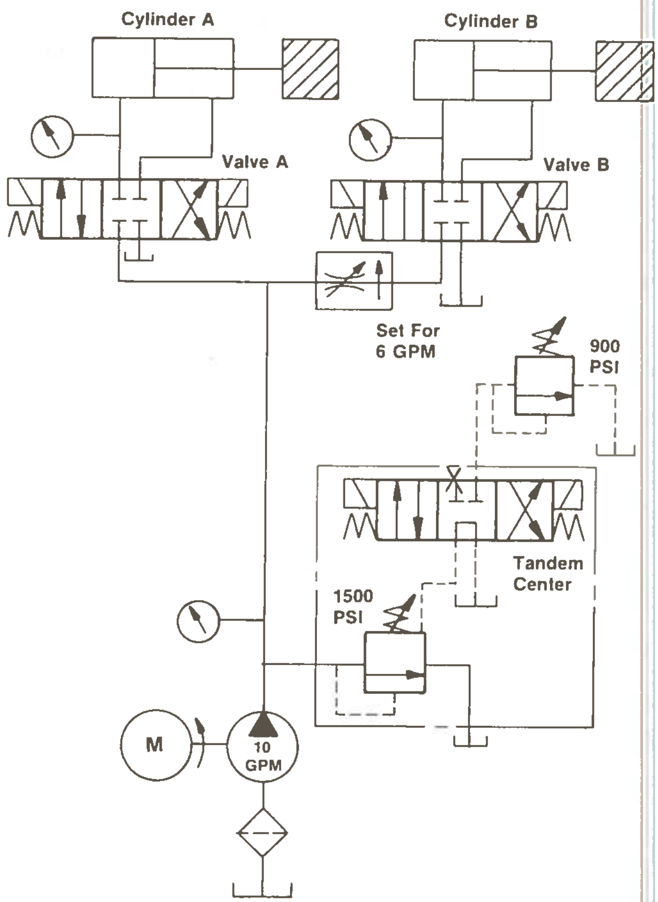
If the relief directional valve were equipped with a closed center, three maximum working pressures could be selected. (Figure 7-24)

Referring to our two cylinder circuit, assume that both cylinders retract separately and that the flow rate entering the rod side of cylinder B is required to be no more than 6 GPM (22.7 LPM). With flow control adjusted to meter 6 GPM (22.7 LPM) into the cylinder, excess pump flow would have to be dumped over the relief valve at the relatively high setting of 900 PSI (62 bar) when retract pressure is 50 PSI (3.4 bar). This generates unnecessary pressure resulting in heat.

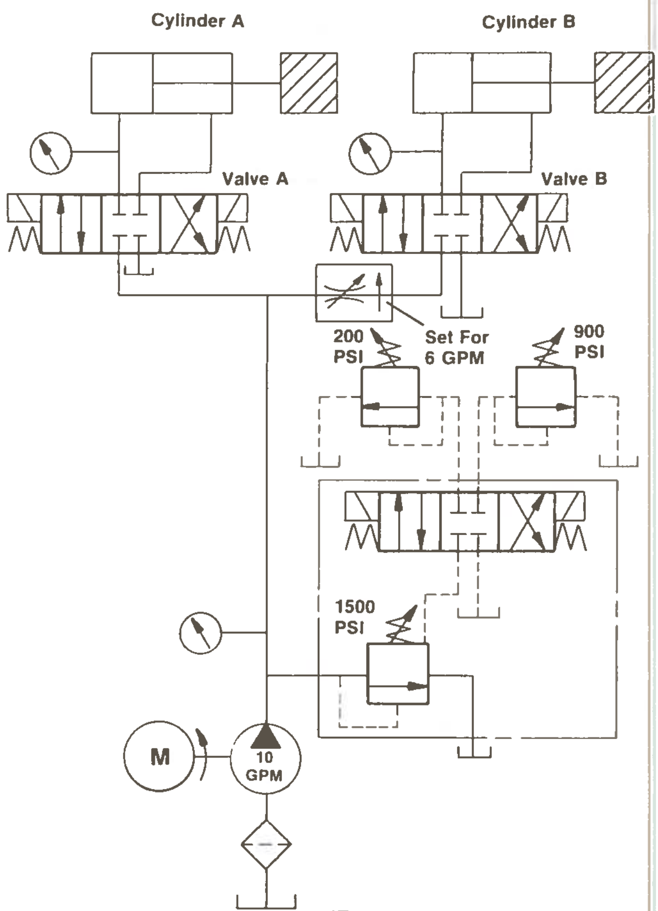
Incorporating another remote pilot valve adjusted for 200 PSI (13.8 bar), the relief directional valve could connect main spring chamber with this low pressure setting during cylinder B retraction. Power generation could then more evenly be matched with actuator output during this portion of the cycle.

In (Figure 7-25) a 15 GPM (56.8 LPM) pump's maximum pressure is controlled by a pilot operated relief valve with two remote pilots and a vent. The directional valve controls which pilot is in control. The following chart can be constructed about the circuit (Figure 7-26).

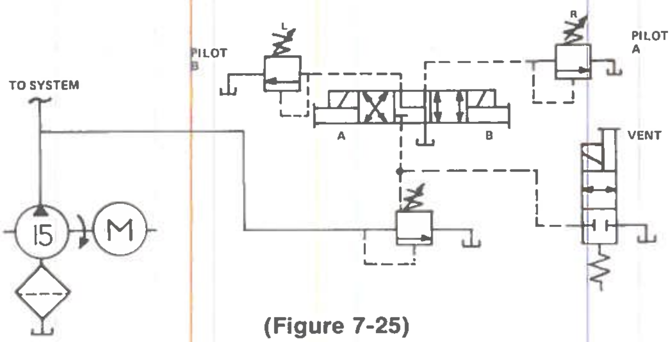
The relief has a low vent spring. Its rated flow is 30 GPM (113.6 LPM)



(Figure 7-23)



(Figure 7-24)

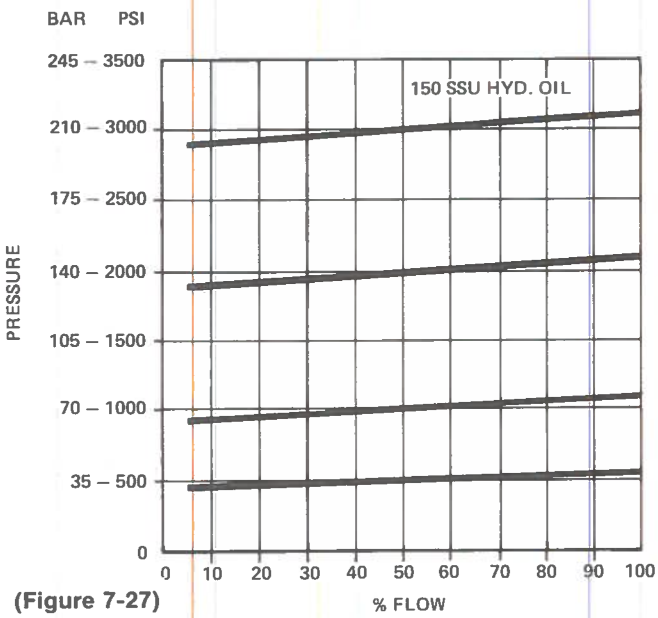


(Figure 7-25)

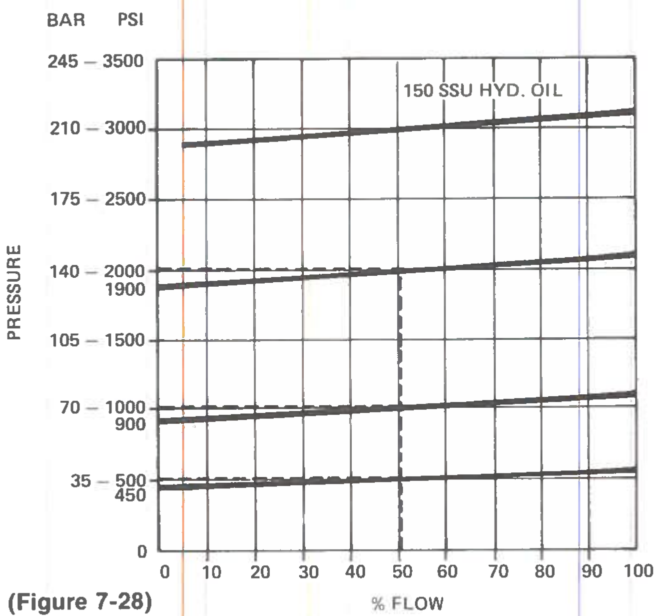
Solenoid Energized	Pilot In Control
None	Internal
A	Pilot A
B	Pilot B
Vent	Vent

Pilot	Relief Valve Setting PSI/Bars
Internal	2000/138
Pilot A	1000/69
Pilot B	500/34.5

(Figure 7-26)



(Figure 7-27)



(Figure 7-28)

1. Find the cracking pressure with:
   a. no solenoids energized
   b. solenoid A energized
   c. solenoid B energized
2. What is the pressure drop when the vent solenoid is energized?
3. What is the heat generation rate when the relief is vented?

SOLUTION:

To find the pressure at which the valve starts to open at various relief valve settings, catalog data must be examined. The data is shown in (Figure 7-27).

Entering the graph at 50% flow, one can find the cracking pressure at each point. (Figure 7-28)

Opening pressure at:

1. a. 1900 psi (131 bar)
   b. 900 psi (62 bar)
   c. 450 psi (31 bar)
2. To find the pressure drop when the valve is vented, a new graph must be entered.

The given information states that a low vent spring is present. This new chart is entered at 15 GPM (113.6 LPM) line and extended upward to the low vent line. At this point, a horizontal line is drawn and the vent pressure (vent pressure is system pressure) is read directly. (Figure 7-29)

The pressure of the system with the vent energized is 45 PSI.

3. The heat generation rate in the system can be found through the use of the power equation.

Heat Generation Rate = Flow x Pressure x .000583

= 15 GPM x 45 PSI x .000583

= .4 HP (.3 KW)

Without the relief valve vented we find:

Heat Generation Rate = Flow x Pressure x .000583

= 15 GPM x 2000 PSI x .000583

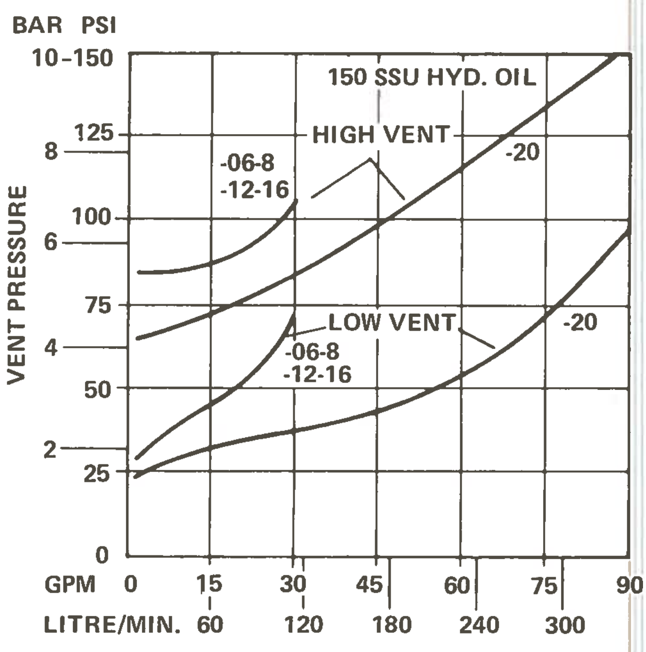
= 17.5 HP for 15 sec. (13 KW)

This would mean an increase in generated heat rate of about 45 times. And remember, this heat must be dissipated either through a heat exchanger or a reservoir. It should be intuitive that not venting the relief valve in this case is an example of poor energy conservation.

Relief valves are one of the large power consumers if not applied thoughtfully. When using them, the following check points should be applied:

1. Match the relief valve setting to the system setting. Good engineering practice dictates at least a 150 psi (10.3 bar) pressure differential between relief and system pressure.
2. When the system is in operation with no other pressure or flow control valves in the circuit, make sure the valve is not opening.
3. When possible, vent the relief for the idle portion of the machine cycle.

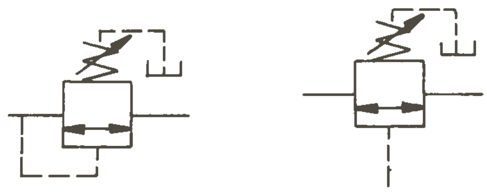
Now by taking a fundamentally equivalent valve and adding an external drain, we have what is hydraulically called a sequence valve.



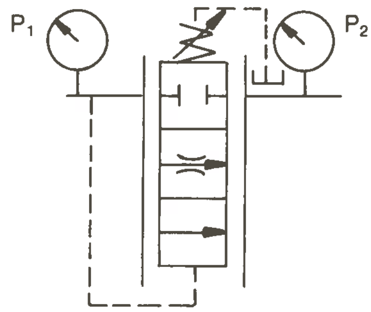
(Figure 7-29)

SEQUENCE VALVE

A sequence valve is a normally closed or non-passing pressure control valve. Its symbol is shown in (Figure 7-30). A sequence valve is placed in a circuit to cause actions to take place in a specified order. This valve will also maintain a setpoint pressure setting in its primary line, as long as flow exists in its secondary leg. A typical application for such a valve is in a clamp-drill circuit. Here clamping must take place first, and a minimum pressure must be present before the drill may proceed. This valve is typically internally operated and externally drained.

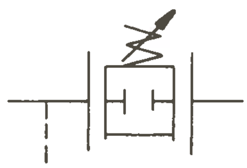

(Figure 7-30)

In order to get a better understanding of the operation of this valve, lets expand the symbol shown in (Figure 7-30) to the drawing shown in (Figure 7-31). As one can see, the valve is still shown internally piloted and externally drained.



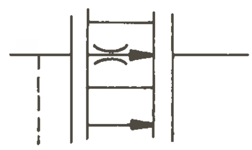
(Figure 7-31)

When the pressure is at P_1, 100 psi (6.9 bar) lower than the spring setting, the valve will typically stay closed as it is pictured. (Figure 7-32)


(Figure 7-32)

As the pressure at P_1 operates in the band 100 psi below to 100 psi (6.9 bar) above the setting of the valve, the intermediate position will come into play.

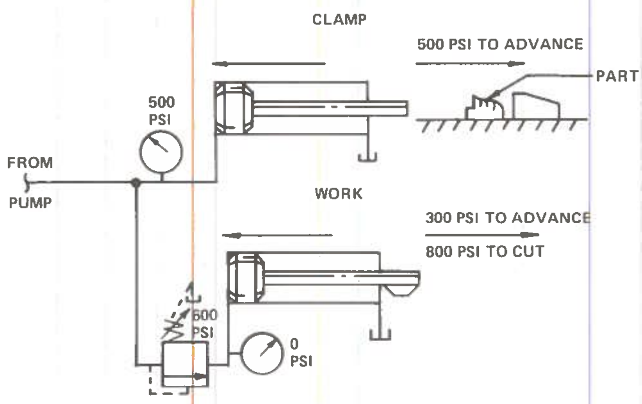
This intermediate position will exist as long as P_1 is in the said band and P_2 is less than the spring setting. The pressure drop through the valve will be equal to P_1 minus P_2. (Figure 7-33)


(Figure 7-33)

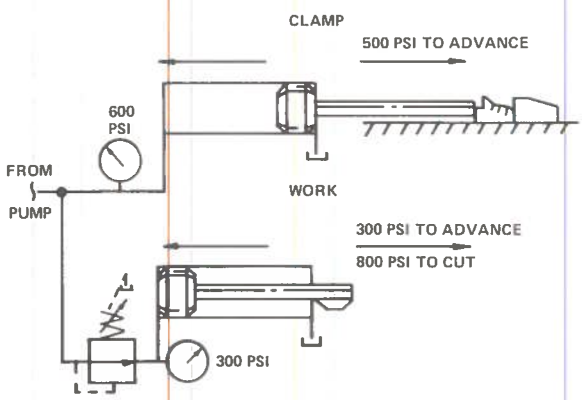
If P_2 should exceed the spring setting by 100 PSI (6.9 bar) the third position will come into play P_1 will also remain higher than the spring setting.)

It is during this stage of operation that the pressure drop through the valve is a function of the fluid and the design of the valve. An approximate figure for

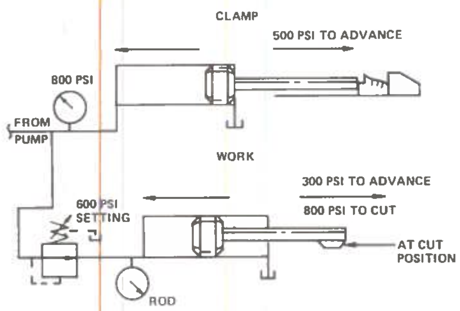
pressure drop can be obtained as per the discussion of unloading valves in Chapter 6.



(Figure 7-34)



(Figure 7-35)



(Figure 7-36)

THE SEQUENCE VALVE IN A CIRCUIT

The sequence valve, as its name implies, is used to ensure that a predetermined sequence will occur. An example is shown in (Figure 7-34).

In this circuit, 500 psi (34.5 bar) is needed to extend the clamp. While clamping pressure must be of at least 500 psi (34.5 bar) it may increase as high as 1000 psi (69 bar) without causing any problems. The work cylinder extends at 300 psi (20.7 bar) until it begins to cut. When cutting begins, the pressure increases to 800 psi (55 bar). Cutting exists for ⅓ of the total cylinder stroke.

It can be seen that without the sequence valve, that the work cylinder would extend until it begins to cut. At this point, the part would not be clamped causing possible injury to personnel and machinery. With the sequence valve set higher than the clamp pressure, the clamp extends at 500 psi (34.5 bar). The valve remains in position (1) during this period of time.

When the clamp contacts the part, the pressure in the system will begin to rise. Once it reaches 600 psi (41 bar), the sequence valve will open and pass full pump flow.

To pass flow during this part of the cycle, the valve spool moves to psition (2). This position causes the valve to act as a restriction. The pressure at the primary port is 600 psi (41 bar) and the secondary pressure is 300 (20.6 bar). This means that 300 psi (20.6 bar) is dropped across the valve and sensed as heat. This is fairly constant throughout the advance portion of the work cylinder.

When the work cylinder begins the cutting mode, the system pressure increases to 800 psi (55 bar) (Figure 7-36). This pressure at primary and secondary ports is approximately the same and equal to 800 psi (55 bar). This is because the valve has shifted to position (3) with the pressure drop as a function of the fluid and the design of the valves flow paths. The valve will remain in this position through the rest of the work stroke.

The heat generation rate can be substantial. To illustrate this, consider the following example.

EXAMPLE: (7-3)

Find the heat generation rate for the sequence valve when:

1. the clamp is extending

2. the work is advancing
3. the work is cutting

Consider 20 GPM to be flowing through the valve when working.

SOLUTION:

a. In this condition, the valve is in position (1). No flow exists therefore, the heat generated is zero.

b. The valve is now passing flow. The pressure drop through the valve is 300 psi (20.6 bar).

Heat Generation Rate = GPM x PSI x .000583 (5-1)
Heat Generation Rate = 20 x 300 x .000583
Heat Generation Rate = 3.5 HP (2.6 KW)

c. The valve is in position (3) where the pressure drop is minimal. From the procedure outlined previously in Chapter 6, an approximate pressure drop would be about 90 PSI (6.2 bar)

Heat Generation Rate = 20 x 90 x .000583 (5-1)
Heat Generation Rate = 1.04 HP (.81 KW)

From the previous example, it can be seen that when the spool shifts to position (2), the pressure drop becomes high. It is during this time that careful consideration should be given to its application.

DIRECTLY OPERATED COUNTERBALANCE VALVE

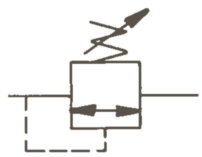
The counterbalance valve is a normally closed valve similar in construction to a relief valve. Its major purpose is to prevent a hydraulically suspended weight from falling uncontrollably when a directional valve is shifted. The symbol for the valve is shown in (Figure 7-37).

The expanded version of the valve is also given in (Figure 7-38). It shows two positions and is functionally equivalent to the directly operated relief valve.

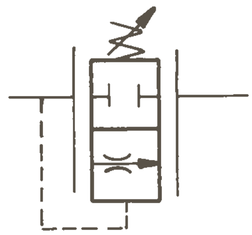
An example of the counterbalance valve in a circuit is shown in (Figure 7-39). It is an example of a press circuit.

Without the use of the counterbalance valve, the weight of the platen will cause it to fall uncontrollably when the directional control valve is shifted to extend the press. This happens because as flow is directed to the capend of the cylinder, it will not be able to fill the cylinder due to the excessive rate of speed of the platen.

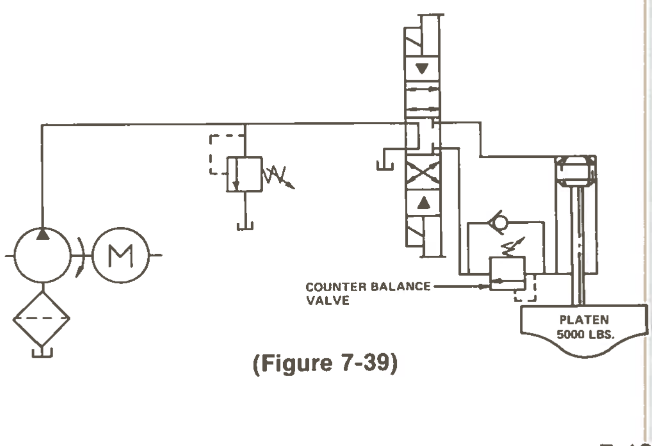
By adding the counterbalance valve just downstream of the head end cylinder port, a back



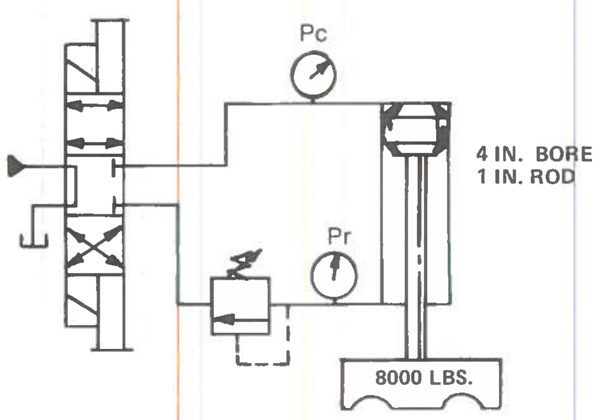
(Figure 7-37)



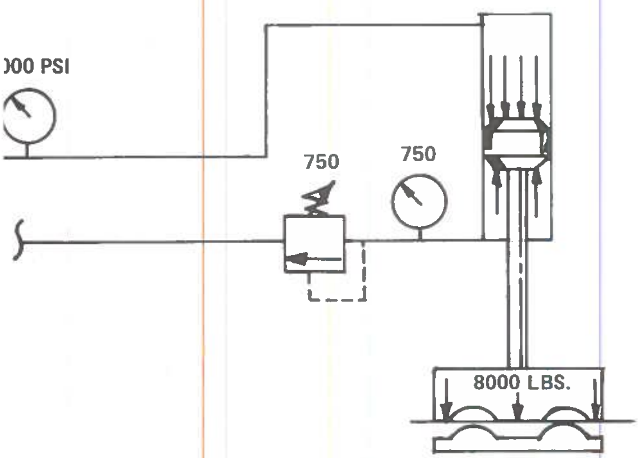
(Figure 7-38)



(Figure 7-39)



(Figure 7-40)



(Figure 7-41)

pressure is created acting on the rod end area. By adjusting this valve, the correct pressure can be obtained to keep the platen from falling. In other words, the weight of the platen and any tools it is supporting is "counterbalanced" throughout its stroke by the back pressure in the rod cavity.

EXAMPLE: (7-4) (Figure 7-40)

In the circuit shown, find the:

a. minimum setting of the counterbalance valve

b. the pressure at Pc to make the cylinder extend

c. the heat generation rate when the counterbalance valve is passing flow (Flow into the cylinder) 5 GPM (18.9 LPM)

d. the maximum force developed by the press (The relief is set for 1000 psi (69 bar)

SOLUTION:

a. The pressure at Pr working on the annulus area must be capable of suspending 8000# load (35584 N). The annulus area is:

$$\text{Annulus area} = Ae\ (in^2) = Ap - Ar = \frac{\pi\,(dp^2 - dr^2)}{4}$$

The pressure can be found by rearranging the equation:

Ft (#f) = Pr (psi) x Ae (in²) - Pc x Ap

$$Pr = \frac{Ft}{Ae} = \frac{8000\#f}{11.78\ in^2} = 680\ \text{PSI (47 bar)}$$

In order to be on the safe side, the valve should be set for about 750 psi (52 bar).

b. The pressure needed to make the cylinder extend can be found from (2-5a) NOTE: when the valve passes flow, the pressure at the head end will be equal to the setting of the valve.

Ft (#f) = Pr (psi) x Ae (in^2) - Pc (psi) x Ap (in^2) (2-5a)

8000 = 8835 - Pc (12.56)

Pc x (12.56) = 835#

Pc = 66.5 psi (4.6 bar)

c. The heat generation rate will be:

Heat generation rate = GPM x PSI x .000583 (5-1)

The flow rate into the cylinder is given to be 5 GPM. The flow rate out can be found by using equation (2-6a)

$$\text{Rod velocity} = V = \frac{Q\ in \times 231}{Ap\ (in^2)} = \frac{Q\ out \times 231}{Ae\ (in^2)}$$

$$= \frac{5\ GPM \times 231}{12.56} = \frac{Q\ out \times 231}{11.78}$$

Q out = 4.69 GPM (17.7 LPM)

The heat generation rate can now be found.

Heat generation rate = 4.69 GPM x 750 PSI x .000583 (5-1)

= 2.05 Hp (1.5 KW)

d. The force developed by the press can be a tricky calculation. When the platen reaches the metal to press, the diagram looks like (Figure 7-41).

The force developed in a pressing operation is one of compression. We can then apply equation (2-4a) which reads:

$FC = Pc\ (psi) \times Ap\ (in^2) - Pr\ (psi)\ \ Ae\ (in^2)$ (2-4a)

This equation must be modified due to the platen weight. Because the platen causes the rod to be in tension, its weight must be subtracted from the force side of the equation.

$Fc - \text{platen weight} = Pc \times Ap - Pr \times Ae$ (7-1)

$Fc = Pc \times Ap - Pr \times Ae + 8000$

$= 1000 \times 12.56 - 750 \times 11.78 + 8000$

$Fc = 12560 - 8835 + 8000 = 11725\#$

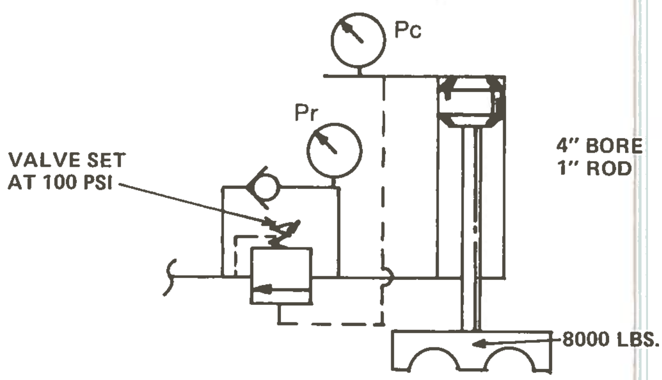
It can be seen from the equation that the pressure developed by the counterbalance valve subtracts from the total force that can be developed during the pressing operation. As a matter of fact, the counterbalance valve seems to nullify the weight of the platen. To eliminate this, a remotely operated counterbalance valve can be used.

REMOTELY OPERATED COUNTERBALANCE VALVE

With a remotely operated counterbalance valve, the expanded diagram of the valve contains three positions. The first two positions are the same as those found in directly operated counterbalance valves. However, the third position is a fully open position. The valve will enter this position when the pressure in the pilot pressure exceeds the spring setting by 100 psi (6.9 bar), independent of the weight of the platen.

Considering the circuit shown in (Figure 7-40) let's replace the directly operated valve with a remotely operated counterbalance valve. (Figure 7-42).

With remote operation, the valve remains in position one, restraining the load from falling. The gage Pr will read the same value as before — Pr = 680 psi (47 bar). When flow is introduced into the cap end of the



(Figure 7-42)

cylinder, the platen is still balanced on its downward stroke. If the platen attemps to pull away from the fluid flow, pressure will drop in the cap end as well as the pilot line. The valve will close and allow the flow to catch up. The pressure at Pc will be equal to the spring setting, therefore;

Pc = 100 psi (6.9 bar)

The pressure at Pr can be calculated from equation (2-5a)

Ft = Pr x Ae - Pc x Ac (2-5a)

8000 = Pr x 11.78 - 100 x 12.56

8000 + 1256 = Pr x 11.78

$$Pr = \frac{9256}{11.78} = 785 \text{ psi } (54 \text{ bar})$$

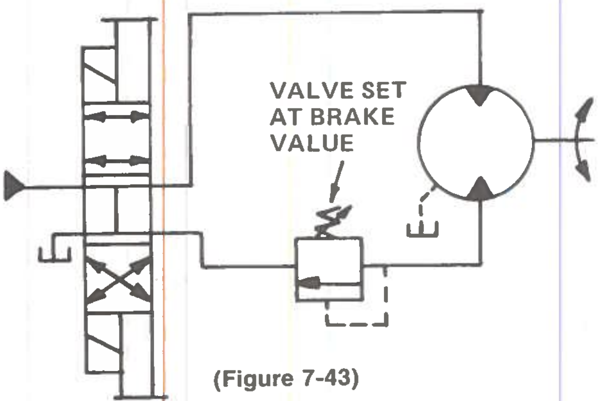
When the platen reaches the material at the die, the pressure in the cap end of the cylinder will begin to build. As soon as the cap pressure exceeds 100 psi (6.9 bar), the pressure in the rod end Pr will go to 0. In other words, the valve has moved to its third mode of operation. The pressing force can now be calculated using equation (7-1).

Fc = platen weight + Pc x Ap - Pr x Ae

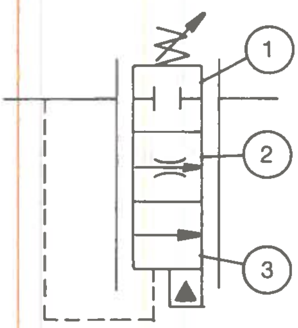
Fc = 1000 x 12.56 - 0 x 11.78 + 8000

Fc = 20,560 #f (91450N)

NOTE: It should be kept in mind that in specialized applications using remotely operated counterbalance valves, the force acting on the rod may not be constant. If this is the case, an eratic motion will exist as the rod extends. If this is the case, a directly operated valve will diminish this type of problem.



(Figure 7-43)



(Figure 7-44)

BRAKE VALVES

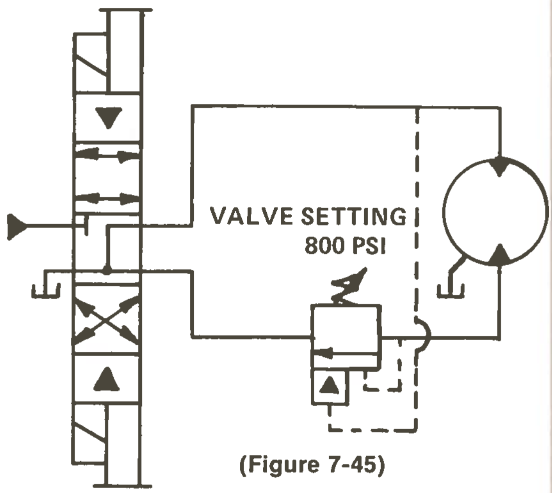
On a motor circuit a direct operated counterbalance valve may be used to control a runaway tendency of a spinning load. As per (Figure 7-43) consider the valve being set for 800 psi (55.2 bar). This will present a back pressure to keep the load from running away from the pump flow. However, it will also have this 800 psi (55.2 bar) more than the work load pressure. For example, if the motor accepts 10 GPM, (37.8 LPM) 4.6 Hp (3.4 KW) would be wasted in this application. This disadvantage can be overcome by using a brake valve.

A brake valve is a marriage between a directly and remotely operated valve. It has both types of pilots. The expanded version of the valve is shown in (Figure 7-44).

The internal pilot works exactly as it does in a

counterbalance valve. When the pressure at the pilot exceeds the equivalent spring setting, the valve moves to position (2) where the flow is restricted causing a back pressure to exist. Should the pilot pressure fall below this setting, the valve will move to position (1) and close. The direct pilot can never cause the spool to move to position (3).

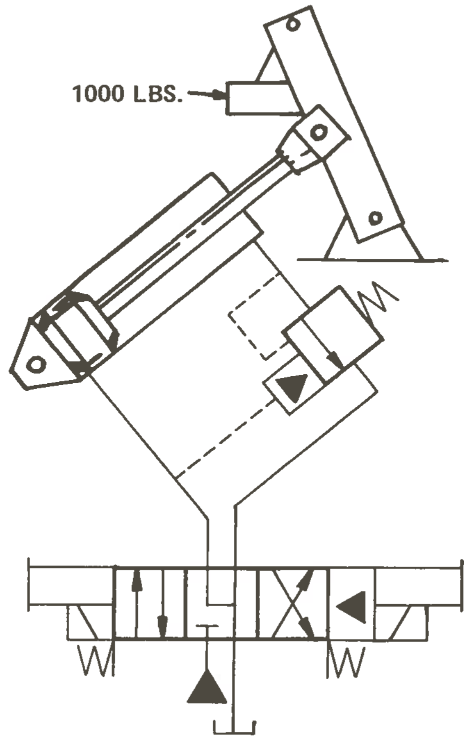
When the remote pilot is pressurized, the operation of the valve is different. The valve is constructed so that a force multiplication factor is built into the remotely operated pilot section. This multiplication factor lets the spool shift with less pressure than the directly operated pilot. The multiplication factor for this valve is many times greater than 5. For instance if its pilot ratio was 8 and the spring setting was 800 psi (55.2 bar), the remote pilot pressure needed to shift to position (3) would be about 800/8 or 100 psi (6.9 bar). The need for two pilots will become more evident when the valve is placed in a circuit. Lets use the example valve and put it in a circuit.



(Figure 7-45)

With the brake valve set for 800 psi (Figure 7-45) the valve will open when 100 psi is present in the motor inlet line because the remote pilot is connected to this line. This means that as long as the load pressure is greater than 100 psi (6.9 bar), the valve spool will remain shifted to position (3) and cause little resistance to flow. However, if the load was allowed to runaway, pressure would drop in the motor inlet line, (below 100 psi (6.9 bar) and not reopen until a back pressure of 800 psi (55.12 bar) was sensed at the direct pilot connected at the outlet of the motor. This back pressure at the motor outlet would then be used to slow down the load.

An overcenter valve is similar to a brake valve but force multiplication factors less than 5 at the remote pilot. This ratio gives a smoother motion as a cylinder load crosses over the center of an arc. (Figure 7-46)



(Figure 7-46)

CONCLUSION

In Chapter 7, we dealt with the basic pressure controls: relief sequence, counterbalance and brake valves. Careful consideration should be given to applying these valves to a circuit. If applied hapharzardly, large pressure drops may occur which will be sensed as heat. Because of this, the heat dissipation characteristics of the hydraulic system must be considered to ensure that no excessive temperature rise is present. These valves should be selected to pass the system flow without restricting it severely.

CHAPTER 8

FLOW CONTROLS

When the speed of a cylinder or motor is greater or more varied than desired, the most commonly applied device to correct this malfunction is a flow control device — a restriction in the flow path. With this restriction, less flow will enter the cylinder or motor. In positive displacement systems, the remaining flow will pass through the relief valve. In pressure compensated variable volume systems, the pump will destroke to deliver the desired volume. However, to better understand the workings of a flow control, let us first look at the flow equation.

FLOW EQUATION FOR LIQUIDS

The equation for flow of a liquid is quite complex and an explanation of that formula would be more academic than useful. Because of this, we will look at the flow equation that describes the flow through a simple circular restriction.

IT IS: $Q = Kdo^2 \sqrt{\frac{\Delta P}{\rho}}$

(8-1)

Where Q is in (in^3/min)

Where K is a function of valve type

Where do^2 is the orifice area (in^2)

Where ΔP is the pressure drop (psi)

Where ρ is the density of the fluid

By examining the equation, we find that there are three parameters that may govern the rate of flow. They are:

1. do . . . which is related to the adjustment level of the circular restriction.
2. ΔP which is related to the pressure drop across the circular restriction.
3. ρ which is the density. If the temperature changes, the density and viscosity of the fluid will change, and the effective aperture may also change.

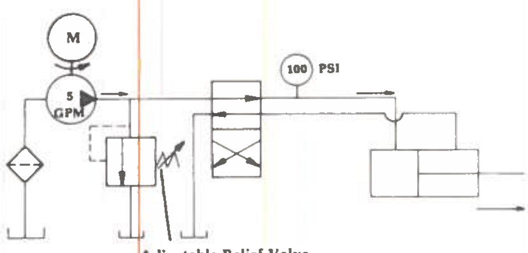
By relating these three points to basic flow controls, an understanding of why many types of these devices exist in the market place.

CHANGING PARAMETERS IN A CIRCUIT

Let us say a flow control is inserted in the following circuit, to slow the cylinder from 385 in/min (0.16m/s) to 132 in/min (0.055 m/s).

The original circuit is shown in (Figure 8-1).

When the piston is moving, the pressure generated in the system is 100 psi (6.9 bar). With pump flow



(Figure 8-1)

equal to 5 GPM and 3 in² (19.35 cm²) area piston, the rod speed is:

$$V = \frac{Q}{A} \times 231 = 385 \text{ in/min } (0.16 \text{ m/s})$$

(2-6a)

A flow control is needed because a speed of 132 in/min must be maintained. The circuit changes are shown in (Figure 8-2) The flow control setting is:

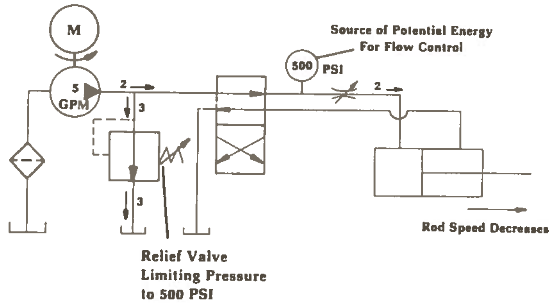
$$Q = \frac{V \times A}{231} = \frac{132 \times 3}{231} = 1.7 \text{ GPM } (6.48 \text{ LPM})$$

The flow control is set for 1.7 GPM (6.48 LPM). As the pump begins to work, it sees the flow control as a restriction. Since the flow control can only pass 1.7 GPM (6.48 LPM) and the pump must displace 5 GPM (18.9 LPM), the system pressure begins to build. When it reaches about 400 psi (27.6 bar), the relief valve begins to open. It is at a pressure about this that it directs 3.3 GPM (12.42 LPM) of the pump flow through the relief valve. This is a characteristic of all fixed displacement pump systems.

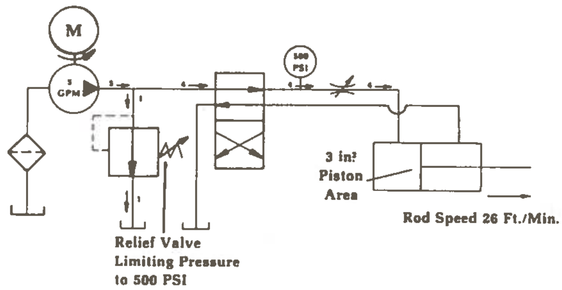
Now, if we change the setting on the flow control to 4 GPM (15.1 LPM), the piston speed will increase to 308 in/min (.13 m/s) and only 1 GPM (3.78 LPM) will flow across the relief valve, as in (Figure 8-3). It is quite evident any change in the size of the orifice will affect the flow rate through a flow control. Now let us look how point two, the pressure drop, can affect the flow.

Considering circuit (Figure 8-4), let us now say that the load is not constant. As a matter of fact, it can require anywhere between 100 psi to 300 psi (6.9 to 20.7 bar) to sustain motion. (Figure 8-4) When the load pressure is 100 psi (6.9 bar) the rod speed will be 240 in/min (0.10 m/sec). Now if the load pressure increases, the flow rate through the flow control valve will decrease. This is due to the fact that less potential energy is available to push the flow across the valve. In other words, flow across a fixed restriction is proportional to the available pressure differential across the restriction. If the pressure differential or drawing force is decreased, flow decreases. The converse is also true. Because of this, when the load pressure equals 300 (20.7 bar), the flow across the flow control may very well drop to about 3 GPM (11.3 LPM). The same phenomenon would happen if the relief valve setting was changed. The point to be considered here is when the load varies, flow across a restrictor type flow control will also vary.

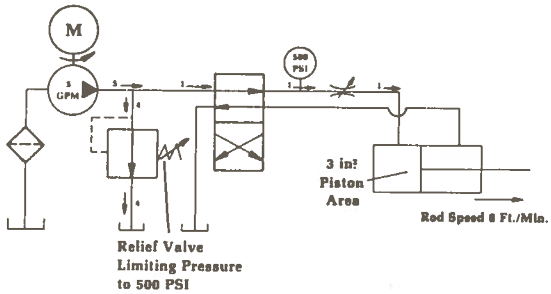
The final parameter to consider is viscosity. The reason why viscosity changes may not be completely evident. Viscosity is related to the temperature. As



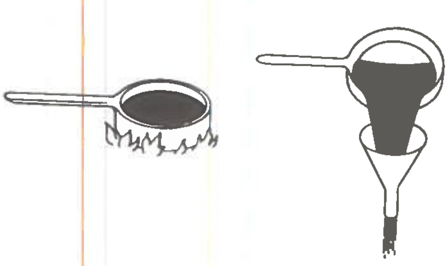
(Figure 8-2)



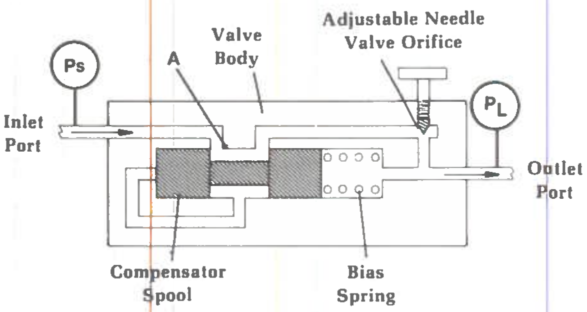
(Figure 8-3)



(Figure 8-4)

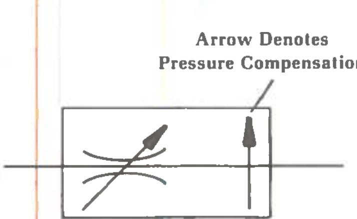


(Figure 8-5)



Pressure Compensated Flow
Control Valve - Restrictor Type

(Figure 8-6)



Pressure Compensated
Flow Control Valve
Symbol

(Figure 8-7)

the temperature of the oil increases, viscosity decreases and vice-versa.

It is realistic to say that it is easier to push a less viscous fluid through a pipe as it is easier to pour water through a funnel than it is molasses. And just as molasses is easier to pour when it is hot, oil will flow more easily due to a lowering of its viscosity. If (Figure 8-5) is considered, when the temperature is low, as at startup, the flow rate is as shown. However, as the oil heats up as the day progresses, its viscosity may drop and the flow through the valve may increase.

We have shown here what causes flow to change in a system. If control is not precise, a needle valve (which controls only the orifice annulus) may be sufficient. However, if additional accuracy is needed with varying loads and cycling temperatures, modifications of the basic needle valve are necessary. Let us first modify the valve to correct for variable pressure.

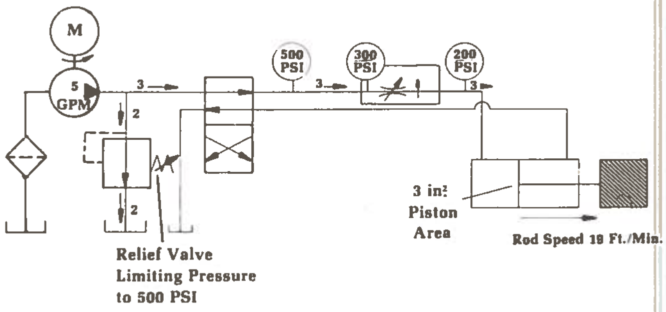
PRESSURE COMPENSATED FLOW CONTROL

In order to maintain constant flow, the pressure differential across the constant restriction must be held constant — ie. pressure changes must be neutralized across the orifice. This can be done by adding a spool which senses both up and down stream pressures in order to keep the differential pressure across the orifice constant. A pictorial drawing of such a valve is shown in (Figure 8-6).

This valve must maintain constant flow no matter if Ps or PL change. First, consider PL, the load pressure, increasing. A change of this nature when applied to a needle valve will cause the flow rate to drop. In this valve, this increase in pressure will cause the spool to move to the left, making path A larger. In this way, the flow remains relatively the same. If the pressure Ps, the system pressure would increase, the spool would move to the right so that the pressure ahead of the orifice would remain constant. As you probably have already noted, a bias spring is present in this type of valve. It is this bias spring that keeps the spool shifted to the left at start up and keeps it from shutting off due to pressure surges. Its value is anywhere between 100 and 150 psi (6.9 and 10.35 bar). This means that a drop of a least this amount must be taken across the valve. If a pressure drop of less than 100 psi (6.9 bar), this particular valve will not maintain its pressure compensation abilities.

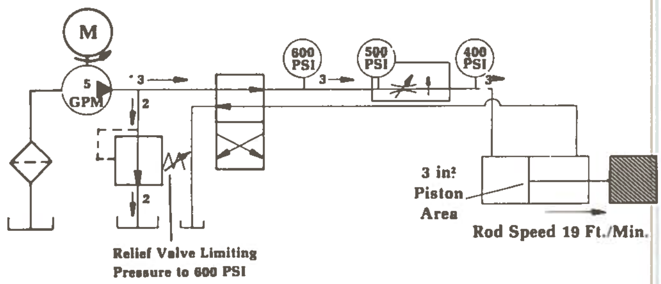
PRESSURE COMPENSATED FLOW CONTROLS IN A CIRCUIT

Using the same system as in (Figure 8-3) and substituting a pressure compensated flow control (Figure 8-7), (Figure 8-8) is obtained. The relief valve is set for 500 psi (34.5 bar) and is constant; the flow control is set for 3 GPM (11.3 LPM). The pump attempts to push its total flow of 5 GPM (18.9 LPM) through the flow orifice. Since this is impossible, the relief valve opens and lets the remaining flow pass across the relief. This valve will maintain a value of 3 GPM (11.3 LPM) as long as the load pressure is 100 psi (6 9 bar) below the pressure at the pump. Should the load pressure exceed this value, the spool in the flow control valve will shift to the left causing pressure compensation to cease. It should be kept in mind that the excess pressure dropped across a flow control valve is sensed as heat.



(Figure 8-8)

If the relief valve was reset to 600 psi, (41.4 bar) the flow control would still maintain 3 GPM (11.3 LPM) (Figure 8-9). The final parameter which must be compensated for is that of temperature. This can be obtained when using a temperature compensated flow control.



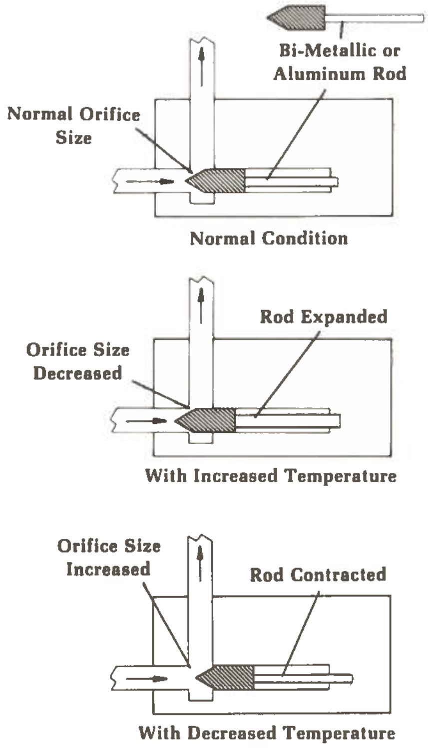
(Figure 8-9)

TEMPERATURE COMPENSATED FLOW CONTROL

One method of temperature compensation is the use of the principle of different coefficients of thermal expansion — or an aluminum rod in a cast iron valve body. The rod is attached to the movable section of a variable orifice and controls the size of the orifice with a change in temperature increases. The heat expands the rod which pushes the movable section of the orifice toward its seat decreasing the opening. The flow rate for the heated fluid through the smaller orifice is the same as the flow rate through the normal orifice before heating. Consequently, flow rate is not affected by an increase in temperature. (Figure 8-10)

If temperature is decreased, flow rate tends to become less. The decreasing temperature causes the rod to contract which pulls the movable section away from its seat, increasing the opening.

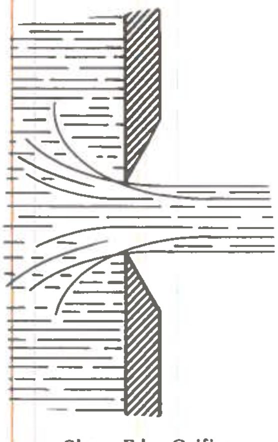
The flow rate for the cooled fluid with the larger orifice is the same as the flow rate through the normal orifice before it was cooled. Therefore, flow is not affected by a decrease in temperature.



(Figure 8-10)

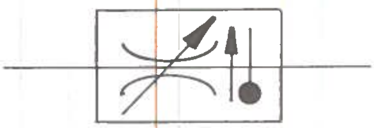
Another method for temepraturе compensation is through the use of a sharp edge orifice. (Figure 8-11)

Laboratory experiment has shown that when liquid passes through a properly shaped orifice with a



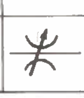
Sharp Edge Orifice

(Figure 8-11)

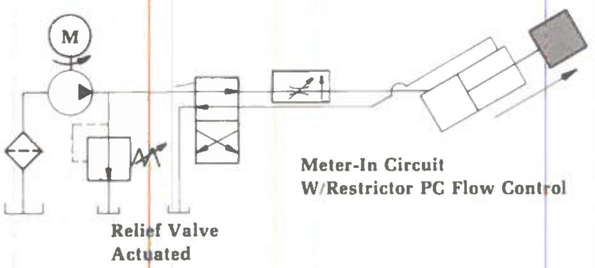


Temperature - Pressure Compensated Flow Control Valve Symbol

(Figure 8-12)

	Symbol	Terminology	Flow Will Vary With		
			Control Setting	Pressure	Temperature
Type 1		temperature pressure compensated flow control	YES	±2 to 4% 2900 psi (200 bar) change	±1% 80 to 140°F (26 to 60°c)
Type 2		pressure compensated flow control	YES	±5% for a 2900 psi (200 bar) change	YES
Type 3		needle valve	YES	YES	YES

(Figure 8-13)



(Figure 8-14)

sharp edge, rate of flow is not affected by viscosity (and therefore temperature). The manner in which liquid is sheared, while moving across a sharp edge, is of such a character that it actually cancels out or neutralizes the effect of a fluid's viscosity. This is due to the fact that a little of the fluid is stopped at the wall of the sharp edge.

Temperature compensated flow controls, by themselves, do not exist. If temperature compensation is needed, a pressure-temperature compensated flow control must be purchased. (Figure 8-12)

With this valve, flow will remain within certain limits over a range of temperatures and pressures. Catalog data should be consulted for the limitations of particular valves.

FLOW CONTROL TYPE SUMMARY

To sum up, there are three types of flow control: (Figure 8-13)

Through the use of this chart, one can obtain the correct flow control for any application. It should be apparent that the way a valve is placed in a circuit has a lot to do with how much total heat will be generated. The three common configurations for placing a flow control in a circuit are: meter in, meter off, or bleed off. Let us discuss each of these configurations in detail.

CYLINDER APPLICATIONS METER IN CIRCUIT

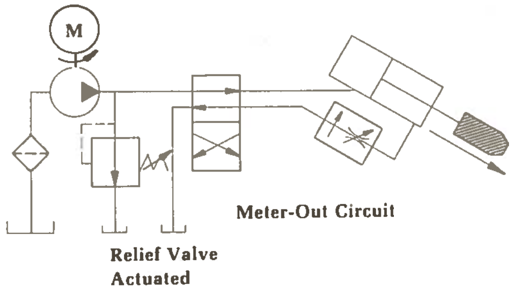
A meter in circuit is one which controls the rate in which fluid enters a cylinder or motor. It is generally used with the load keeping the rod in constant compression. (Figure 8-14) If, for example the load would attempt to put a tensile load on the rod, the load would run away; the pump flow not keeping up with the actuator speed. This type of circuitry is commonly used when lifting a weight vertically.

METER OUT CIRCUIT

In many applications, the force exerted on the rod can change from compression to tension. An example of this would be a load passing over the center of an arc where the load pressure would suddenly drop to zero. Another case a drill bit breaking through a part. As the tip breaks through, it tends to grab and pull at the bit. A metered out circuit will prevent this from happening.

A flow control valve placed at the outlet port of an actuator controls the rate of flow exiting the actuator. This gives positive speed control to actuators used in drilling, sawing, boring and

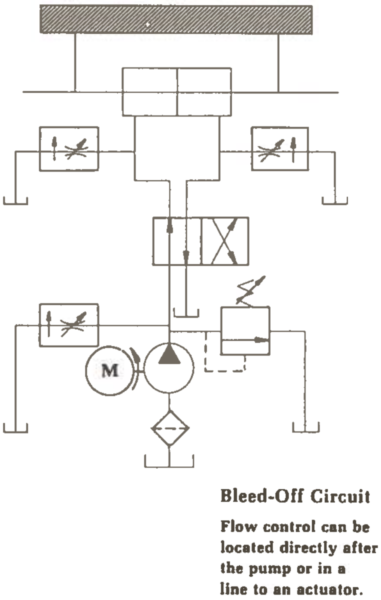
dumping operations. This is the most popular industrial flow control application found. (care should be taken when light loads are applied to a cylinder that excessive intensification does not occur) (Figure 8-15)



(Figure 8-15)

BLEED OFF CIRCUIT

Another type of flow control circuit is the bleed off circuit. Here, whatever flow is undesirable, it is returned back to tank at the existing system pressure. This means that the flow control does not cause any added restriction to the pump. Besides generating less heat, the bleed off circuit can also be more economical than a meter in or meter out. For instance, a 100 GPM (378.5 LPM) flow is delivered by the pump and only 80 (302.8) is needed to the system. In the meter in circuit, a 80 GPM (302.8 LPM) flow control is needed; 60 GPM (227.1 LPM) for the metered out case (Depending on the cylinder sized) and only 20 GPM (75.7 LPM) for the bleed off. (Figure 8-16)

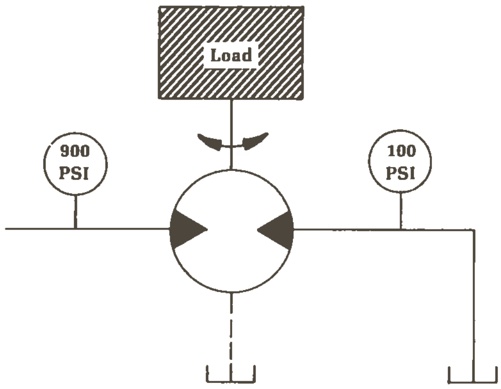


(Figure 8-16)

Even with these apparent advantages, the bleed off is not a very popular means of control. This is due to the many facts. One is that the bleed off will precisely control only the flow to tank. If any leakage rate changes in the system, the speed of the actuator will change. This leakage is directly related to the wearing of components, along with the increase in leakage due to the volumetric efficiencies of the pump caused by pressure fluctuations from operating load levels. Also, an apparent danger may exist if the load at the actuator would go to zero. If this would be the case, little, if any, of the flow would go across the bleed off flow control. This means that the speed of the actuator may increase greatly.

SPEED CONTROL OF MOTORS

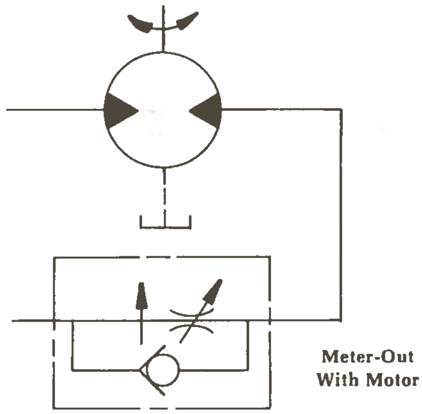
To accurately control the speed of a hydraulic motor, a meter out circuit is used. This, of course, assumes that the motor is externally drained. (Figure 8-17)



(Figure 8-17)

Hydraulic motors are generally externally drained. This means a portion of the flow entering the motor ends up as leakage. As the torque requirement and the pressure at the motor increases, more flow runs out the drain. As a result, motor shaft speed decreases. (Figure 8-18)

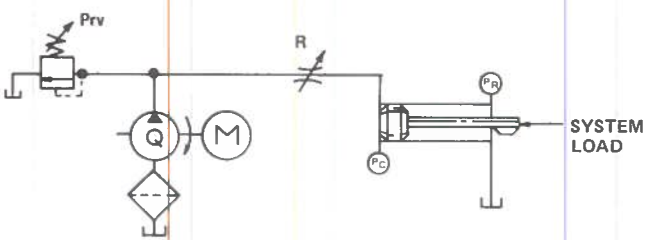
A meter out circuit controls the flow as it discharges from the motor and is not concerned with leakage. This is the only circuit which can control a motor's shaft speed accurately regardless of load.



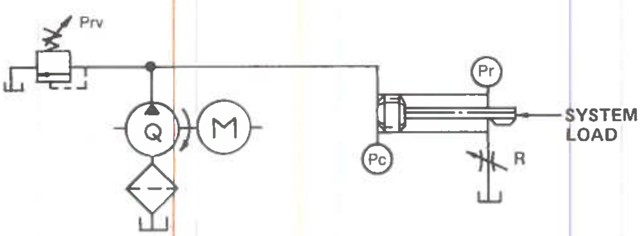
(Figure 8-18)

Meter In
Meter Out
Bleed Off

(Figure 8-19)



(Figure 8-20)



(Figure 8-21)

HEAT GENERATION RATE DUE TO FLOW CONTROLS

The heat generation rate in a system employing flow controls depends greatly on the system design. However, for a fixed displacement pump in a system with flow controls, a bleed off circuit will always be more efficient. However, safety is a big consideration if bleed off is used. Care should be exercised when employing a bleed off. For a variable displacement pressure compensated pump network, the meter in or meter out may be more efficient depending on the working system flow rates. Heat generation for a metered in and metered out will always be the same.

To calculate the heat generation rate in the system, it is a matter of summing up pressures and flow rates on a system. The efficiency of various flow control placements will be discussed first. (Figure 8-19)

In a meter in circuit with a fixed displacement pump, part of the flow is directed across the relief valve. The pump flow is Q, the system load is Pc, the relief pressure Prv and the flow rate setting is R. The pump efficiency (Ep) is assumed to be 80% above 100 psi. The heat generation rate in the system is (neglecting pump inefficiency): (Figure 8-20)

Heat Generation Rate = heat across flow control + heat across relief

Heat eneration Rate = flow across flow control (R) x (ΔP) x K + flow excess x relief valve setting x K

Heat Generation Rate =
$$(R \times (Prv - Pc) \times K) + (Q-R) \times (Prv) \times K$$

Heat Generation Rate =
$$K\,(R \times Prv - R \times Pc + Q \times Prv - Prv \times R)$$

Heat Generation Rate = (Q x Prv - R x Pc) x K
(8-1)

It can be seen from the above equation that as Q x Prv approaches R x Pc the heat rate goes to zero. The efficiency (Eo) of the meter in is:

$$\%\ \text{efficiency (Eo)} = \frac{HP\ out}{HP\ in} = \frac{\dfrac{Pc \times R \times K \times 100}{PRV \times Q \times K}}{Ep} = \frac{Pc \times R \times Ep \times 100}{Prv \times Q}$$

(8-2)

For a meter out system, the same type of equations may be developed. First, let us relate the flow rate into the cylinder to the flow rate out of the cylinder (Q out). From Chapter 2, we may find: (Figure 8-21)

$$\text{Rod speed} = V = \frac{Q \times 231}{A} \qquad (2\text{-}6a)$$

Since the rod speed is constant the ratio of Q in to Q out is:

$$V = \frac{Qin \times 231}{Ap} = \frac{Qout \times 231}{Ae}$$

Since Qout = R and $Ap = \frac{\pi\, dp^2}{4}$ $\quad Ae = \frac{\pi\,(dp^2 - dr^2)}{4}$

$$Q\ in = \frac{R \times dp^2}{dp^2 - dr^2}$$

Heat Generation Rate = heat across the relief + heat across flow control

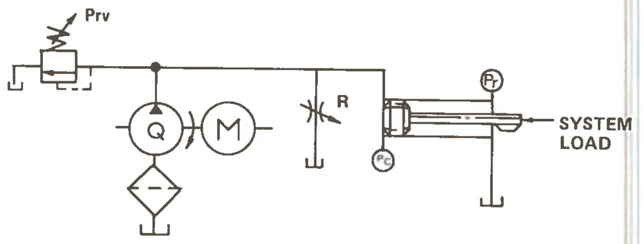
Heat Rate =((Qpump - Qin) x Prv x K) + (R x Pr x K)

Since $Q in = \frac{R \times dp^2}{(dp^2 - dr^2)}$

$$\text{Heat Rate} = \left[\left(Q\ pump - \frac{(R \times dp^2)}{(dp^2 - dr^2)}\right) \times Prv + (R \times Pr)\right] \times K$$

The efficiency of the circuit is:

$$\%\ Efficiency\ (Eo) = \frac{Hp\ out}{Hp\ in} = \frac{(Prv \times \frac{Ap}{Ae} - Pr) \times R \times K}{\frac{Q \times Prv \times K}{Ep}} = \frac{(Prv \times \frac{Ap}{Ae} - Pr) \times R \times Ep \times 100}{Q \times Prv}$$



(Figure 8-22)

In a bleed off circuit, the relief valve does not open. Therefore, the only heat generated is equal to that across the flow control. (Figure 8-22)

The efficiency of this circuit is:

$$\%\ Efficiency\ (Eo) = \frac{Hp\ out}{Hp\ in} = \frac{(Q-R) \times Pc \times K \times 100}{\frac{Q \times Pc \times K}{Ep}} = \frac{(Q-R) \times Ep \times 100}{Q}$$

Let us now look at a few examples of the use of flow controls in a system.

EXAMPLE: (8-1)

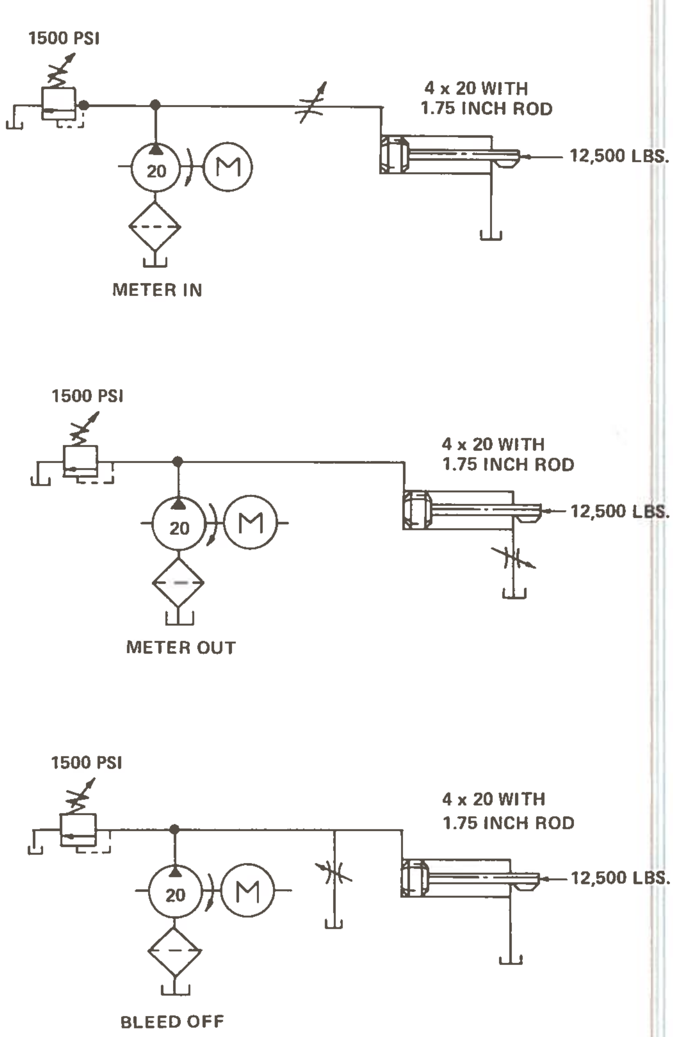
A fixed displacement pump delivers oil at a rate of 20 GPM (75.7 LPM). A rod speed of 275 in/min (0.116 m/s) is required. Find the approximate pressures, flows and heat generation rate due to the flow control (neglecting pump efficiency) and also efficiencies of the following circuits including pump efficiencies. Pump efficiency above 200 psi (13.8 bar) is to be an assumed 80%. (Figure 8-23)

SOLUTION:

First, we have to find the needed flow rate into the cylinder. This can be obtained from the formula in Chapter 2.

$$Q = \frac{V \times A}{231} \qquad (2\text{-}6a)$$

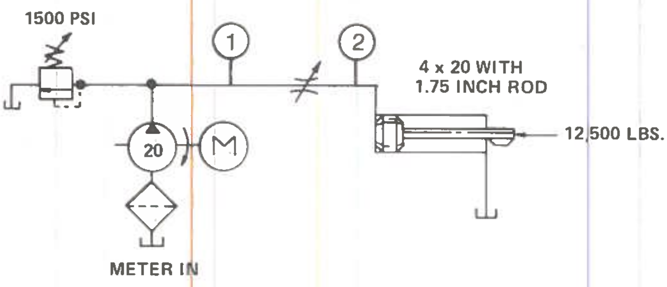
Where A is the piston area $= \frac{\pi\, 4^2}{4} = 12.56\ in^2\ (81\ cm^2)$



(Figure 8-23)

$$Q = \frac{275 \times 12.56}{231} = 15 \text{ GPM } (56.7 \text{ LPM})$$

This means the flow rate setting for the meter-in flow control is 15 GPM (56.8 LPM). Let us now find the pressure needed to solve the problem. (Figure 8-24)



(Figure 8-24)

Since the flow rate of the pump is 20 GPM (75.7 LPM) and the flow control will only pass 15 GPM (56.8 LPM), 5 GPM (18.9 LPM) of the flow must pass through the relief valve. The pressure at this point in the system (gage 1) would be near the relief valve setting. Let us assume it is 1500 psi (103.5 bar).

Gage 2 (Pc) would read whatever pressure is needed to overcome the resistance of the load. Using the equation from Chapter 2, gage 2 pressure can be found.

$Fc = Pc \times Ap - Pr \times Ae$ (2-4a) Since Pr is associated to be $Fc = Pc \times Ap$

or

$$Pc = \frac{Fc}{Ap} = \frac{12560}{\frac{\pi\ 4^2}{4}} = 1000 \text{ psi } (69 \text{ bar})$$

Heat generation rate is found via equation (8-1) and efficiency equation (8-2).

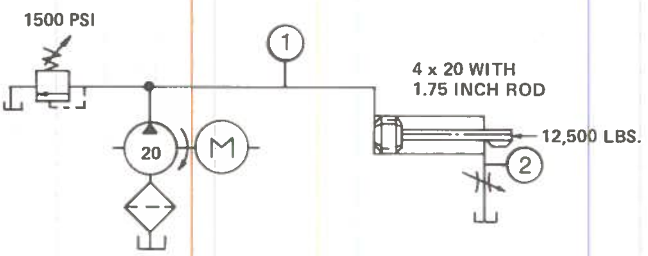
Heat Rate = (Q x Prv - R x Pc) x K

= (20 x 1500 - 15 x 1000) x .000583

= 8.75 HP (6.52 KW)

$$\text{Efficiency} = \frac{Pc \times R \times Ep \times 100}{Prv \times Q} = \frac{1000 \times 15 \times .80 \times 100}{1500 \times 20} = 40\%$$

The above heat generation rate does not take into account pump inefficiency.



(Figure 8-25)

For the meter-out system a different flow rate must be determined because of the area difference between the cap and annulus area of a cylinder. The rod speed is to remain constant for all systems.

$$Q = \frac{V \times A}{231} = \frac{275 \times \pi (dp^2 - dr^2)}{4 \times 231} = 12 \text{ GPM } (45.4 \text{ LPM})$$

(2-6a)

The flow rate setting will be 12 GPM (45.4 LPM) (Figure 8-25).

Once again because the system will not accept the entire pump flow, part of the flow is directed across the relief valve. Since the relief valve and cap end of the cylinder are all in the same line, the pressure of this ideal system will be approximately the same. Therefore, we will assume Gage 1 to read 1500 psi. In order to find the pressure at Gage 2, we use the force equation:

$Fc = Pc \times Ac - Pr \times Ae$ (2-4a)

Where $F = 12560$

Where $Pc = 1500$

Where $Ap = 12.56\ in^2$

Where $Ae = 10.15\ in^2$

$12560 = 12.56 \times 1500 - Pr \times 10.15$

$Pr = 619$ psi (42.7 bar)

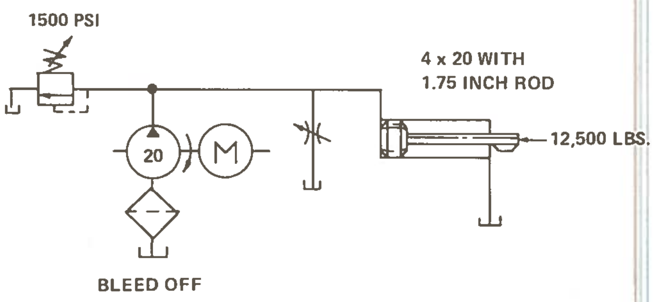
The heat generation rate is:

$$\text{Heat Rate} = \left[\left[\left(Q\ pump - \frac{(R \times dp^2)}{(dp^2 - dr^2)}\right] \times Prv\right] + (R \times Pr)\right] \times K$$

$$= 8.8\ HP\ (6.50\ KW)$$

It should be noted at this time that the heat generation rate and the efficiency of a meter-in and meter-out circuit are equal. Also it should be stated that the accuracy of the meter out is the highest.

For a bleed-off circuit, the pressure is constant fairly constant throughout the system and is therefore assumed to be the case. This means that the pressure between pump and cylinder are equal to the value dicated by the load resistance. (Figure 8-26)



(Figure 8-26)

In this case:

$Fc = Pc \times Ap - Pr \times Ae$ (2-4a)

$12560 = Pc \times 12.56 - 0 \times 10.15$

$Pc = 1000$ psi (69 bar)

The requirement of speed dictates that 15 GPM (56.8 LPM) must enter the cylinder. If the pump expells 20 GPM (75.7 LPM), the flow control must be set to "bleed-off" 5 GPM (18.9 LPM) of the pump flow.

The heat generation rate is:

HEAT RATE = $R \times Pc \times K = 5 \times 1000 \times .000583$

HEAT RATE = 2.9 HP (2.16 KW)

$$\text{Efficiency} = \frac{(Q-R) \times Ep \times 100}{Q} = \frac{(20-5) \times .80 \times 100}{20} = 60\%$$

Summing up what we have determined in the system, we find: (Figure 8-27)

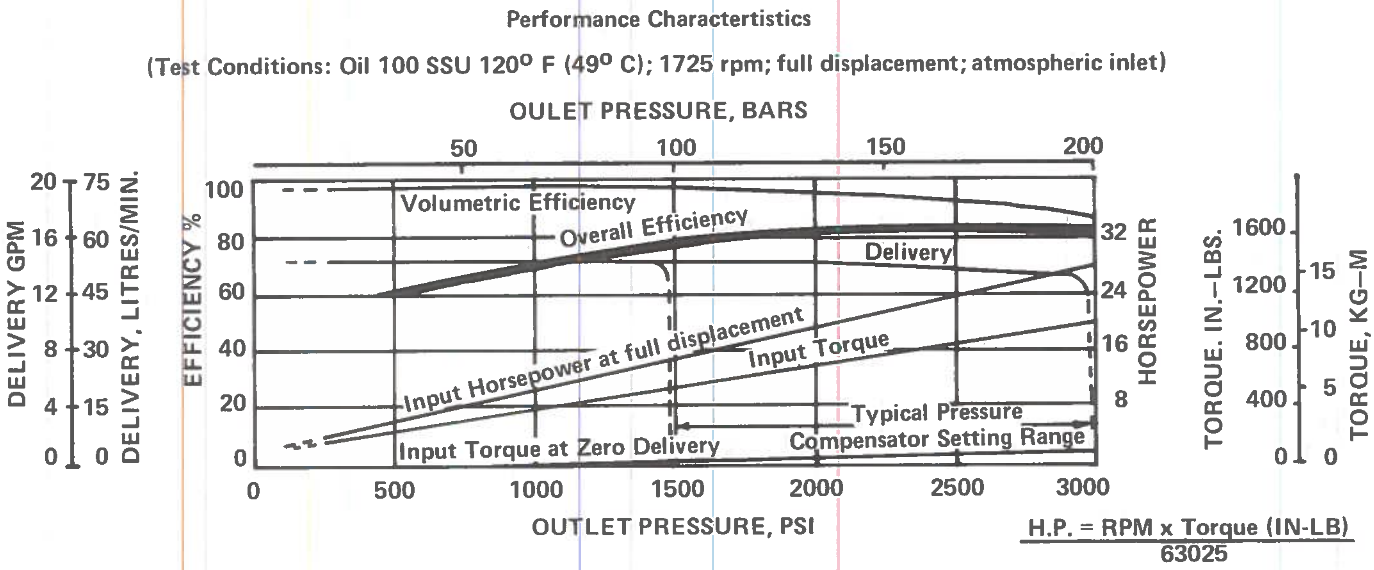
Flow Configuration	Heat (HP/KW) Rate	Efficiency %	Controlability
meter-in	8.75/6.52	40	accurate as long as load keeps incoming oil pressurized.
meter-out	8.75/6.52	40	accurate for all loading whether compressive or tensile.
bleed-off	2.9/2.16	60	least accurate control of velocity. Less certain whether this will actually control speed if load drops to zero.

(Figure 8-27)

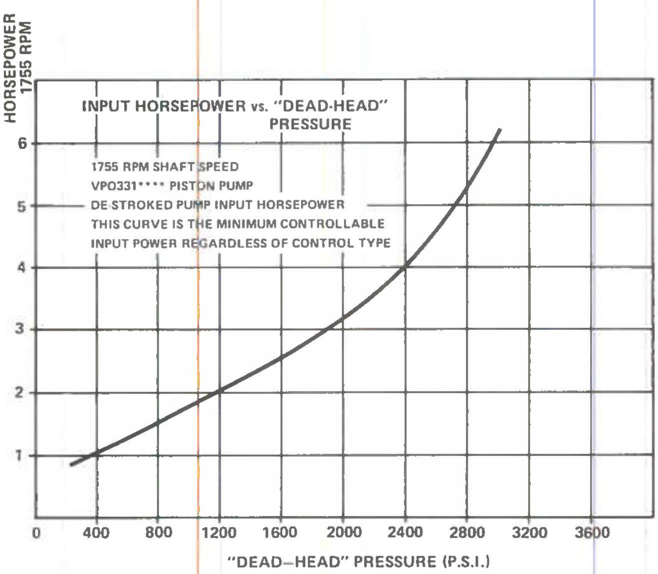
With a variable volume pressure compensated pump, the displacement will vary so that the pump delivers the flow that the circuit requires. This makes it easy to calculate the heat generation rate when a flow control is added to the circuit. But, before we go

any farther, let's refresh ourselves with the characteristics of this type of pump.

For this example, let us consider pump of 14 GPM(53 LPM) variable volume piston pump. Performance characteristics are given in (Figure 8-28). Since the pump will be operating in a compensated mode due to the flow control, data is needed on the input horsepower vs. deadhead pressure. This is used in lieu of the fact that efficiency is not obtainable during this compensation mode.



(Figure 8-28)



(Figure 8-29)

This curve will be used to obtain the circuit efficiency. (Figure 8-29)

First, the heat generation rate across the flow control is a straight forward calculation: (Figure 8-30)

HEAT RATE = flow rate setting of flow control x ΔP x K

HEAT RATE = R x (P-Pc) x K

Where P is the pressure at which the pump delivers needed flow, the efficiency of the circuit is the ratio of power out to power in.

The power out to the system is:

POWER OUT = R x Pc x K

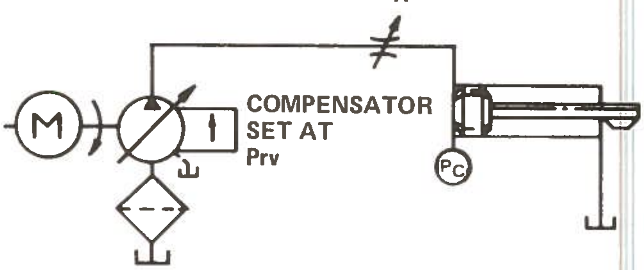
The power into the system from the variable volume pump is a function of flow and full compensation power draw.

Referring to Chapter 4, P Partial Compensation = GPM x PSI x .000583 + Input HP @ full compensation (4-10a).

The efficiency of the system is:

$$\text{Efficiency (Eo)} + \frac{\text{Power out}}{\text{Power in}} = \frac{R \times Pc \times K \times 100}{(GPM) \times PSI \times .000583) + \text{input HP @ full comp.}}$$

(8-9)



(Figure 8-30)

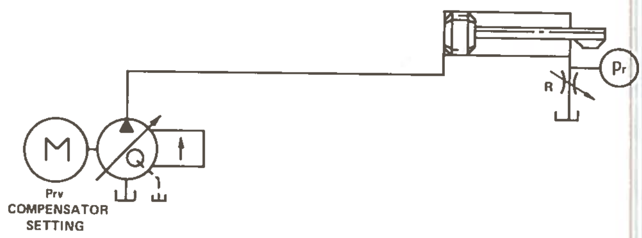
The meter-out system (Figure 8-31) we will have the same amount of heat generation rate across the flow control as was generation rate in the fixed displacement system.

HEAT RATE = Pr x R x K

The efficiency of the system can be calculated by:

$$\text{Efficiency (Eo)} = \frac{\text{Hp out}}{\text{Hp in}} = \frac{\left[(Prv-P) \times \frac{Ap}{Ae} - Pr\right] \times R \times K \times 100}{\left[\frac{Ap}{Ae} \times R \times P \times K + \text{Input Hp @ full compensation}\right]}$$

(8-10)



(Figure 8-31)

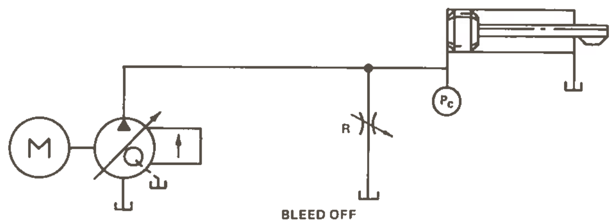
For the bleed-off system (Figure 8-32) the heat generation rate by the flow control is:

HEAT RATE = R x Pc x K

The efficiency of the system would be:

$$\text{Efficiency (Eo)} = \frac{\dfrac{(Q-R) \times Pc \times 100 \times K}{Q \times Pc \times K}}{Ep} = \frac{(Q-R)\,Ep \times 100}{Q}$$

(8-12)



As an example, let us say that the pump whose characteristics are given in (Figure 8-28 and 8-29) is used to power circuit (Figure 8-30, 8-31, and 8-32) with R as 4, 2 and 10 GPM (15.1, 7.52, 37.8 LPM) respectively. The pump is a 14 GPM (53 LPM) pump with the cylinder characteristics of a dp = 2 in (50.8 mm) dr = 1.4 in (35.56 mm). The compensator is set at 1500 PSI (103.5 bar). The load resistance is 2000 pounds (8896 N). Find the heat generation rate across the flow control and the circuit efficiency.

For meter-in, the pressure $Pc = \dfrac{2000}{\frac{\pi 2^2}{4}} = 636$ PSI (43.9 bar)

HEAT RATE = R x (P-Pc) x K

P is read from (Figure 8-29) and is approximately 1475 PSI.

HEAT RATE = 4 x (1475 - 636) x .000583 = 1.95 HP (1.45 KW)

The circuit efficiency is:

$$\text{EFFICIENCY (Eo)} = \frac{R \times Pc \times K \times 100}{R \times P \times K + \text{input HP @ full comp.}} = \frac{4 \times 636 \times .000583 \times 100}{4 \times 175 \times .000583 + 2.4}$$

Where 2.4 HP (1.75 KW) is the power drawn at full compensation.

Eo = 25.4%

For meter-out the heat generation rate is:

HEAT RATE = Pr x R x K

From Chapter 2, $Pr = \frac{Pc \times Ac - Fc}{Ae}$

(2-4a)

$$Pr = \frac{(1475 \times \pi (4/4) - 2000)}{\frac{\pi (2^2 - 1.4^2)}{4}} = 1676 \text{ PSI } (115.6 \text{ bar})$$

HEAT RATE = 1675 x 2 x .000583 = 1.95 HP (1.45 KW)

$$\text{EFFICIENCY (Eo)} = \frac{(1475 \times \frac{\pi\, 4/4}{\pi\, 2/4} - 1676) \times 2 \times .000583 \times 100}{(2(4/2) \times 1475 \times .000583) + 2.4}$$

Eo = 25.4%

For the bleed off:

HEAT RATE = R x Pc x K = 10 x 636 x .000583 = 3.7 HP (2.76 KW)

$$\text{EFFICIENCY (Eo)} = \frac{(4)}{14} \times .6 \times 100 = 17\%$$

Summing up:

	Heat Across Flow Control	Efficiency
Meter in	1.95	25.4
Meter out	1.95	25.4
Bleed off	3.7	17

CONCLUSION

Here is a summary of a flow control's ability to compensate for changing parameters.

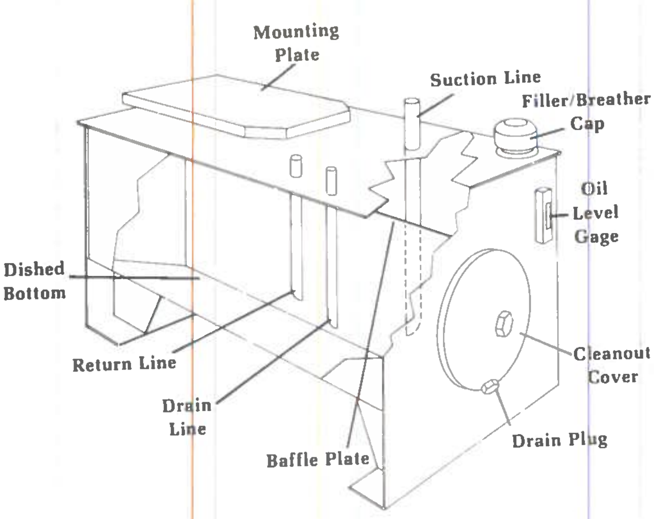
	Symbol	Terminology	Flow Will Vary With		
			Control Setting	Pressure	Temperature
Type 1		temperature pressure compensated flow control	YES	±2 to 4% 2900 psi (200 bar) change	±1% 80 to 140°F (26 to 60°c)
Type 2		pressure compensated flow control	YES	±5% for a 2900 psi (200 bar) change	YES
Type 3		needle valve	YES	YES	YES

When a flow control is added to a circuit in, three different configurations. They are meter in, meter out, and bleed off.

CHAPTER 9

CONDITIONING SYSTEMS
RESERVOIRS AND HEAT EXCHANGERS

Just as a sequence valve is important to the function of an entire machine, the individual conditioning elements are important to the operation of the system. It is these elements: reservoirs, heat exchangers, and filters that keep the hydraulic system functioning as it was designed. We will start our discussion with reservoirs.



(Figure 9-1)

RESERVOIRS

The obvious function of a reservoir is to contain or store a system's hydraulic fluid. However, it's purpose is far more important than just that. The reservoir provides:

a. oil to make up for system leakage

b. a place where fluid born particulate dirt may settle out of the system.

c. a place where entrained bubbles may rise out of the oil.

d. it allows space for the expansion and contraction of the system's volume due to thermal changes.

e. allows the installation of environmental control (heating & cooling) to stabilize a system's temperature.

To make a reservoir more effective, a baffle plate blocks returning oil from going directly to the suction line. This creates a condition that is condusive to particulate dirt settling and entrained air escaping. It also gives sufficient time for the heat in the fluid to dissipate through the reservoir walls.

Fluid baffling and the reduction of aeration is a very important part of proper reservoir operation. For this reason, all lines which return flow should be located below the fluid level, and at the baffle side opposite the power inlet line.

In our analysis of the reservoir, we look for it to meet the following specifications:

a. take care of normal fluid variations in the system (less fluid in system when cylinders are retracted than extended) and make up for longer term fluid losses.

b. take care of thermal requirements - ie - dissipate heat from the working fluid.

c. remove entrained water and keep it separated from the working medium.

d. allow large particulate matter to settle out.

e. prevent ingestion of particulate or liquid contaminants.

In this chapter, we will concern ourselves with "B" because it is very important when analyzing the system. Since pressure losses caused by flow and the hydraulic components generate heat throughout the system, this heat must be dissipated to prevent excess temperature rise. The reservoir must dissipate part of this generated heat.

The quantity of heat the reservoir will dissipate is a function of:

a. its overall size

b. ambient conditions (air temperature, air velocity, exposure to radiant energy)

c. the temperature differential between the fluid and the ambient air.

Heat is transferred by three different means:

1. radiation
2. conduction
3. convection

Radiation is the process by which heat flows from a higher temperature body to a lower temperature body when the bodies are separated in space. This term is generally applied to electromagnetic-wave phenomena. Heat transfer by radiation becomes increasingly more important as the temperature of the radiating body increases. However, the temperatures of industrial reservoirs rarely exceed 160° F. Because of this fact, the effect of radiant cooling on total heat dissipation may be neglected.

Conduction is a process by which heat flows from a high to low temperature body through a medium (solid, liquid or gas) or between mediums of direct contact. This heat from the fluid is transferred to the walls of the reservoir by conduction. Heat also may flow between the reservoir and the floor on which it rests, (wall etc.) by means of their direct contact. However, because there is so little area in direct contact with another surface, conduction heat transfer is also small.

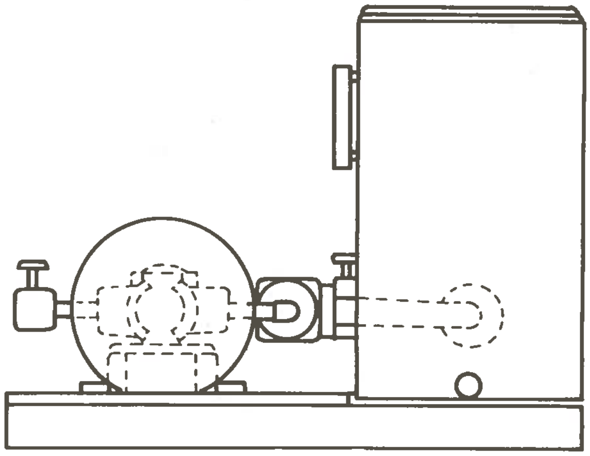
The largest amount of heat transfer available to remove heat from the reservoir is through convection. This is a process by which heat flows from high to low temperature by a combined action of conduction, convection and mixing motion. Convection is the one important mechanism of energy transfer between a solid surface and a liquid or a gas.

Although it is not difficult to work directly with the equations of heat transfer, tables have been constructed to show the heat dissipation characteristics of various size reservoirs. One of these tables



Hydraulic Reservoir Symbol

(Figure 9-2)

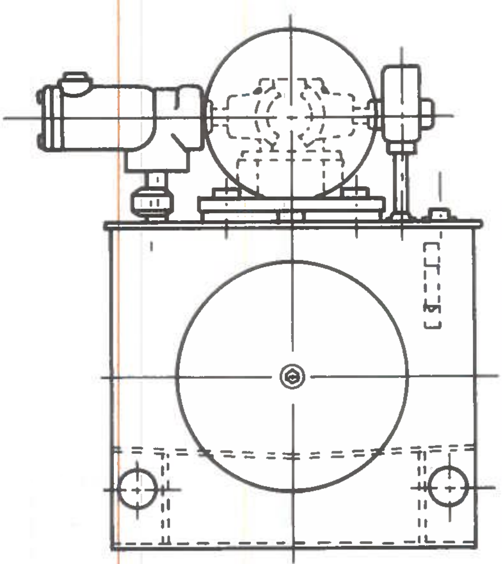


(Figure 9-3a)

RESERVOIR SIZE (GAL.)	H. P. DISSIPATED PER F DIFFERENCE BETWEEN OIL TEMP. AND AMBIENT*
10	0.010
20	0.025
30	0.030
40	0.035
60	0.040
90	0.045
120	0.050
180	0.060
240	0.070

*FOR FULL RESERVOIR WITH PUMP RUNNING AND MILD AIR CIRCULATION.

(Figure 9-3b)



(Figure 9-3c)

(Figure 9-3b) is given. (Figure 9-3b) shows the dimensions of a series of typical reservoirs and their heat dissipation abilities. For instance, a 30 gallon (113.5 L) reservoir will have a dissipation rate of 0.030HP per °F (0.04 KW/°C) difference in temperature. This would mean that if the surrounding temperature was 90°F (32°C) the maximum temperature of the oil was 140°F (60°C), the reservoir's dissipation rate would be:

HEAT DISSIPATION RATE = K x ΔT (9-1)

Where K is reservoir heat dissipation rate constant

Where ΔT is temp. differential

HEAT DISSIPATION RATE = .030 x (140-90) = 1.5 HP (1.1 KW)

It should be kept in mind that an industrial reservoir is usually sized for 2½ to 3 times the pump flow. If the reservoir volume significantly exceeds 3 times the pump flow per minute, the oil may not circulate enough to provide the necessary mixing and heat transfer rate. Hot spots may develop and the given heat dissipation rating of the reservoir will be reduced greatly.

If a reservoir must be sized greater than 3 times the pump flow, a small circulating pump may be installed to keep the oil thoroughly mixed. This will help maintain the entire reservoir at a constant temperature.

EXAMPLE: (9-1)

In a hydraulic system, the heat generated is as follows:

Cycle Part	Heat Generation Rate (HP/KW)	Time (Sec.)
1	.3/ .22	4
2	1.4/1.04	6
3	.6/ .45	3
4	3/2.24	1

The pump is a 15 GPM (56.7 LPM) capacity. Surrounding ambient temperature is 90°F (32°C)

SOLUTION:

The average heat is:

$$\text{HEAT RATE} = \frac{\sum_{n=1}^{\infty} Pn \times tn}{\sum_{n=1}^{\infty} tn}$$

$$\frac{.3 \times 4 + 1.4 \times 6 + .6 \times 3 + 3 \times 1}{14}$$

HP = 1.03 HP (.77 KW)

If we allow the temperature of the oil to be a

maximum of 130°F, (54.4°C), the temeprature differential is 40° F (22.4° C). Applying equation (9-1):

HEAT RATE = K x T (9-1)

1.03 = K x 40

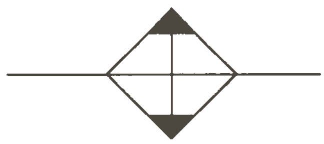
K = 0.026 HP/°F (.035 KW/°C)

This means that if K is greater than or equal to 0.026 HP/°F (0.035 KW/°C) the oil will be within our standards. Since a 30 gallon (113.51) reservoir has a K = .030 HP/°F (0.40 KW/°C) from the standpoint of heat, it will meet the requirements.

Let us now turn our attention to heat exchangers.

HEAT EXCHANGERS (COOLERS)

Heat exchangers are employed in hydraulic systems when the reservoir itself cannot dissipate the generated heat. Two basic types used in this industry use air or water to absorb the heat.



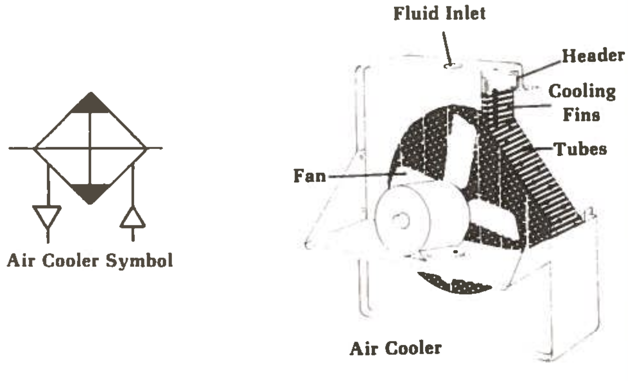
Cooler Symbol

(Figure 9-4)

For economy, manufacturers have developed standard lines of heat exchangers. The selection of standard units requires a thermal analysis to determine which unit of specified size and geometry, can meet the requirements of cooling the hydraulic fluid at a specified rate. With an analysis of this type, the initial cost must be weighed against such factors as life of the equipment, space required, cost of cooling medium, and ease of cleaning and maintenance. It is not uncommon that water types will need cleaning because of deposits left by minerals in the water. Varnish may coat the oil side. Because of this resistance to heat flow varnish may greatly decrease the exchangers overall efficiency. Let us now look at the two types of heat exchangers.

OIL TO AIR EXCHANGER (AIR COOLER)

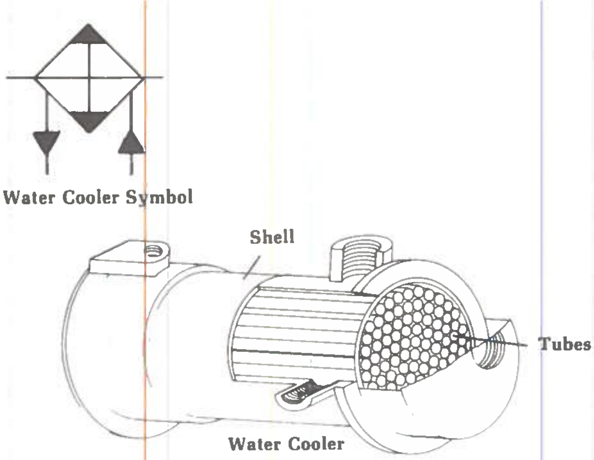
In an air cooler, fluid is pumped through tubes to which fins are attached. To dissipate heat, air is blown over the tubes and fins by a fan. The operation is exactly like an automobile radiator. A similar arrangement may also be used in some applications where natural convection provides the necessary heat transfer. Air directed from the electric motor driving the system's pump is often directed across a fin and tube heat exchanger mounted in the path of such air flow. This is very effective and inexpensive arrangement. (Figure 9-5)



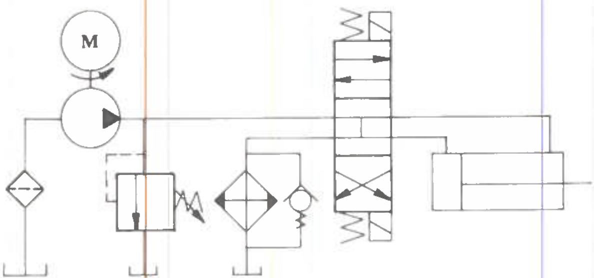
(Figure 9-5)

OIL TO WATER EXCHANGER (WATER COOLER)

A water cooler basically consists of a bundle of tubes encased in a metal shell. In this cooler, a system's hydraulic fluid is usually pumped through



(Figure 9-6)



(Figure 9-7)

the shell and over the tubes which are circulated with cooling water.

This cooler is also known as a shell-and-tube type heat exchanger. Since hydraulic fluid can also be heated with this device by simply running hot water through the tubes, the device may also be used to heat the oil in order to bring it to the proper operating temperature in a cold ambient.(Figure 9-6)

HEAT EXCHANGERS IN A CIRCUIT

Heat exchangers are usually rated at a relatively low operating pressure (150 psi). This requires that they be installed in a low pressure part of the system. If this is not possible, the cooler may be installed in its own separate circulating system often in conjunction with the system filter.

To insure that a pressure surge in a line does not damage a shell and tube type heat exchanger, they are generally piped into a system in parallel with a valve which will prevent excess pressure — ie a check valve with a 65 psi spring.

Typical locations of heat exchangers are: in a system's return line, in the tank line connection of a relief valve, in a case drain line of a variable volume, pressure compensated pump in the return line of an auxiliary filter circuit, etc.(Figure 9-7)

ANALYZING A HEAT EXCHANGER

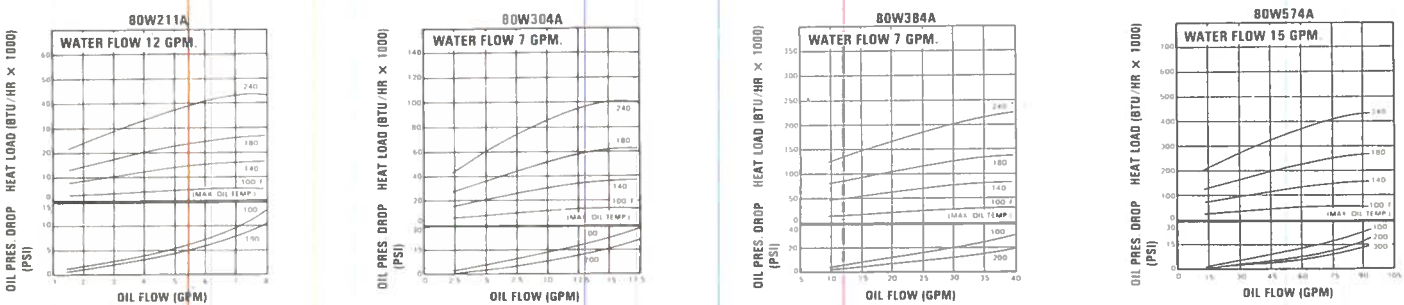
Typical data of heat exchangers is given below in (Figure 9-8). The important data to be considered is the pressure drop due to the heat exchanger and the amount of heat the exchanger will dissipate. Rather than go through a long explanation of the proceeding, let us go directly to a problem to explain the technique.

EXAMPLE: (9-2)

A ten GPM pump generates the following heat rate:

Cycle	Heat Generation (HP/KW)	Time (Sec.)
1	1.6/1.2	4
2	3.1/2.3	2
3	7.3/5.4	6
4	2.2/1.6	10

Select the model, indicated by the number over each graph, which most nearly meets your requirements in terms of water flow, oil flow, oil pressure drop and estimated heat load. All values shown were obtained with SAE 10 oil and a water temperature of 75° (1 HP = 2547 BTU/HR)



(Figure 9-8)

Determine the reservoir size and the heat exchanger from (Figure 9-8) if one is needed.

SOLUTION:

The average heat generated is:

$$\text{HEAT AVERAGE} = \frac{\sum_{n=1}^{\infty} Pn \times tn}{\sum_{n=1}^{\infty} tn}$$

$$= \frac{1.6 \times 4 + 3.1 \times 2 + 7.3 \times 6 + 2.2 \times 10}{22}$$

$$= 3.56 \text{ HP } (2.65 \text{ KW})$$

Let us say that the surrounding temperature (ambient) is 90°F (32°C) and the maximum allowable is 130°F (54°C). K for the reservoir is:

HEAT RATE = K x ΔT (9-1)

3.56 = K x 40

K = 0.089 HP/°F (0.119KW/°C)

Since we do not want to exceed a reservoir size greater than 3 times the pump flow, the largest we can consider is 30 gallons (113.56 litres)

From the previous chart, we find that a 30 gallon (113.56 litre) reservoir has a K = 0.030 HP/°F (.040 KW/°C) which does not meet our thermal requirements. This means that we must install a heat exchanger.

The heat exchanger must dissipate what the reservoir will not. The reservoir will dissipate.

HEAT RATE = K x ΔT = 0.30 x 40 = 1.2 HP (.89 KW)

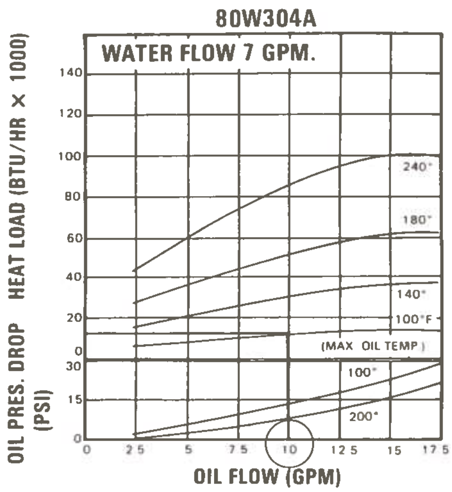
This means we must dissipate.

HEAT EXCHANGER = 3.56 - 1.2 = 2.36 HP (1.76 KW)

Changing horsepower to BTU/hr

Heat Exchanger = 2.36 HP x 2547 = 6011 BTU/hr

Since the pump flow rate has a nominal value of 10 GPM (37.8 LPM), we would select heat exchanger "B". From the curves, this exchanger will dissipate over 10,000 BTU/hr (2.98 KW) with the oil subjected to a temperature of only 100°F (37.7°C)(Figure 9-9)



(Figure 9-9)

CONCLUSION

A reservoir has five major purposes: They are:

1. oil to make up for system leakage
2. a place where fluid born particulate dirt may settle out of the system
3. a place where entrained bubbles may rise out of the oil
4. it allows space for the expansion and contraction of the system's volume due to thermal changes
5. allows the installation of environmental control

(heating and cooling) to stabilize a systems temperature.

One major consideration is how much heat will a reservoir dissipate. The equation developed in this section for heat dissipation is:

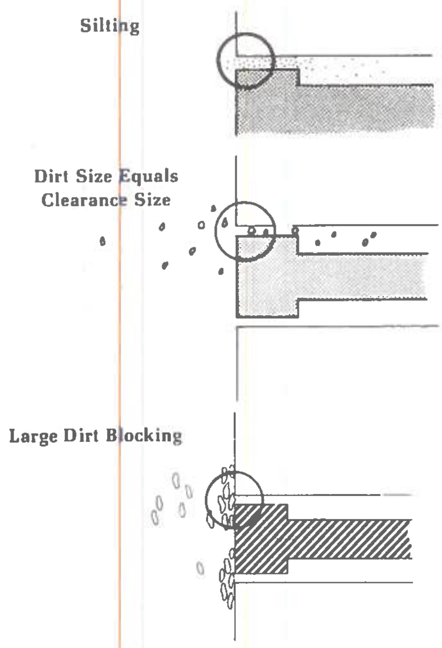
HEAT DISSIPATION RATE = K x ΔT

When all the total cycle heat generation rate cannot be dissipated through system components, a heat exchanger must be added. Two are available: oil to air (air cooler) and oil to water (water cooler). These should be sized for relatively low pressures, making sure that system flow rates are within specified limitations of the heat exchanger manufacturer.

CHAPTER 10

CONDITIONING (FILTERS)

Dirt or contamination is ever present in a hydraulic system. The difference between the contaminant level one system and another is the number of particles of dirt and the size of these particles. Each component with close fitting parts moving relative to one another is sensitive to this dirt, the degree of sensitivity depending on its particular function and design. If a system's filtration needs correspond to the filtration supplied, it will operate for very long periods without showing the usual signs of wear and performance degradation. Experienced maintenance men generally agree that the great majority of component and system premature component failures are caused by dirt.



(Figure 10-1)

TYPES OF DIRT

Probably the greatest problem with dirt in a hydraulic system is that it interferes with lubrication.

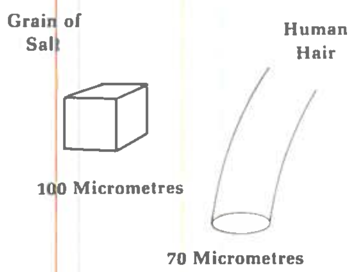
Dirt can be divided into three sizes with respect to a particular component's clearance; that is, dirt which is significantly smaller than the clearance, dirt which is the same size, and dirt which is larger than a clearance. (Figure 10-1)

Extremely fine dirt (silt), which is smaller than a component's clearances may collect in small clearances especially if the contaminant is present in large amounts and the valve is not operated frequently. This blocks or obstructs lubricative flow through the passage.

Dirt which is about the same size as a clearance rubs against moving parts breaking down a fluid's lubricative film. The resulting abrasion creates additional particles of contamination.

Large dirt can also interfere with lubrication by collecting at the entrance to a clearance and blocking fluid flow between moving parts.

A lack of lubrication causes excessive wear, slow response, erratic operation, solenoid burn out, and early component failure.



(Figure 10-2)

HOW PARTICLE SIZE IS MEASURED

One micrometre (micron) is equal to one millionth of a meter or thirty-nine millionths of an inch. A single micrometre is invisible to the unaided eye and is so small that it is extremely difficult to visualize its relative size. To bring the size into perspective, some everyday objects will be defined in terms of the micrometre scale.

An ordinary grain of table salt measures 100 micrometres.

The average diameter of human hair measures 70 micrometres.

Twenty-five micrometres is approximately one thousandth of an inch, the thickness of the paper wrapper on a cigarette.

THE AMOUNT OF DIRT SEEN BY A SYSTEM

The amount of dirt in a system can be directly related to the system's environment. The Basic Fluid Power Research Center (BFPR)* reports that "there seems to be a direct relationship between environment and the system contamination level. In general, the dirtier the environment, the dirtier the system." In a test performed by the BFPR, 44 samples were extracted from 11 hydraulic powered machinery including: 7 farm tractors, 5 loaders, 2 backhoes, dozer, 9 cranes, 3 combines, 1 forklift, 6 various machine tools, 2 hydro-electric plants, 1 dredge and 7 aircraft.

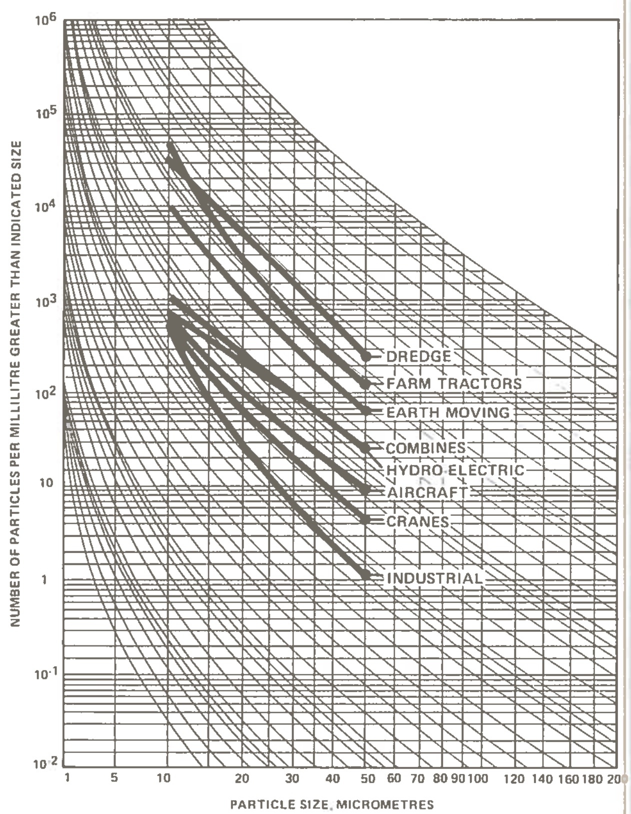
The BFPR deducted that:

"Although the data for each type of equipment appears somewhat scattered, an important result can be seen by calculating the average particle count for each equipment class. These average counts are shown on the Particulate Contamination Chart in (Figure 10-3). The earthmoving class includes loaders, backhoes and dozers, the industrial class includes the machine tools, and the forklift; and the remaining classes include only the single type of machine. A general appreciation of these results can be obtained by studying (Figure 10-3), which indicates that higher contamination levels are usually found in systems which are exposed to dirtier operating environments."

In studying (Figure 10-3), it should become apparent that these typical industrial machines have the cleanest systems.

However, this may be due to a clean industrial environment. Parker has completed some 7000 fluid analysis studies, the major portion of which are machine tool applications. These tests have shown that actual particles distributions closely overlap or exceed the curves shown in (Figure 10-3) for mobile equipment. In other words, industrial systems are as dirty as an average mobile system.

For an average system, the BFPR states that the ingestion rate (number of particles the pump sees) is near 10^8 particles per minute greater than ten micrometres. This is the rate that will be used in further discussions. If the particular environment of a machine is poor, a correction must be applied for additional particle ingression.

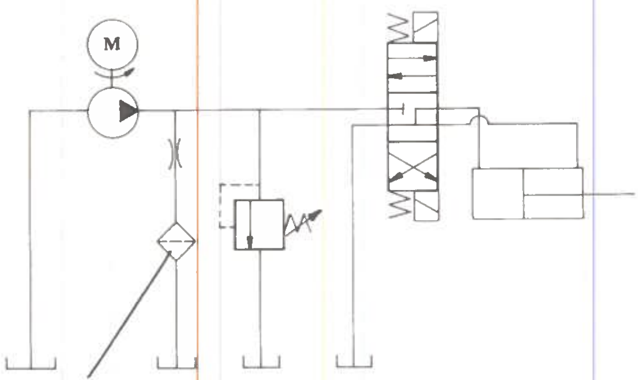


(Figure 10-3)

TYPE OF FILTRATION BY FLOW

In the early days of hydraulics, filtration was considered to be unnecessary, and in many applications this was a correct assumption. Clearances between mating hydraulic components at that time were large and used at lower system pressures as compared to modern standards. Because of these lower pressures, clearances between moving parts could be large, therefore, relatively dirty fluid would not affect opertion that much. Components were dirt tolerant to a large extent.

As system pressures increased, manufacturing processes naturally improved and clearances were reduced. Component operation became much more efficient, but components were less dirt tolerant. Lower component life at increased pressures was recognized, and the need for filtration in relatively simple circuits were realized.



(Figure 10-4)

PROPORTIONAL FLOW FILTRATION

Originally, systems used relatively course filter in the system in such a way that a part of the total volume of a pump was filtered. This was easily done by placing a filter in a system so that a portion of a pump's flow was bled off through the filter. (Figure 10-4)

FULL FLOW FILTRATION

Proportional flow filtration was found to be inadequate after a time, especially as clearances in system components became smaller. Also, large particles or pieces of contaminant caused sudden major failures.

The next step was to place a filter in a system so that all flow to a pump was filtered and where possible, all of the pump's output flow was also filtered through a finer filter.

This full flow filtration is the type of filtration used in most modern hydraulic systems.

TYPE OF FILTRATION BY POSITION IN A SYSTEM

A filter provides protection against particulate contamination causing wear and failure for a hydraulic component. Ideally, each system component would be equipped with its own filter, but this is economically impractical in most cases. The usual practice is to strategically place filters in a system.

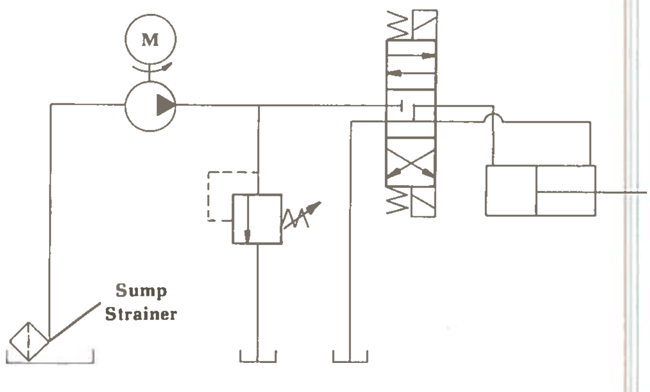
In the majority of applications, the fluid reservoir is a large source of dirt for a system. Since a pump is the heart of a system, as well as one of the most

expensive components in a system, it seems logical that a good place to put a filter is between a reservoir and pump.

SUMP STRAINER

A sump strainer is usually a coarse filter element attached to the end of a pump's suction line, usually deep in the reservoir. (Figure 10-5)

The range of filtration for sump strainers is from perforated metal cylinders with large drilled holes down to 74 micrometres wire mesh. It then protects the pump against damage from large contaminant pieces.



(Figure 10-5)

ADVANTAGES:

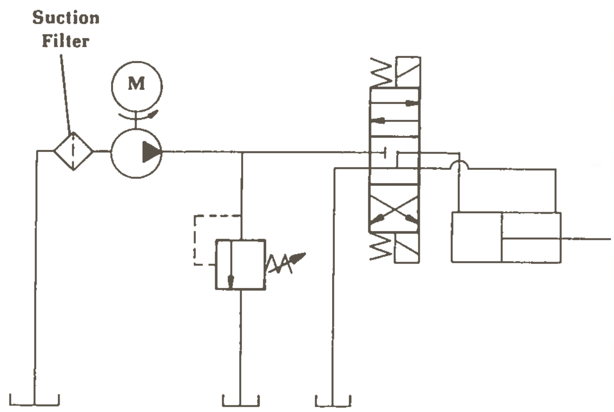
1. Sump strainers protect the pump from large contaminant or foreign matter left in the reservoir.
2. Because they have no filter housing, sump strainers are very inexpensive.

DISADVANTAGES:

1. Being below fluid level, sump strainers are very difficult to service when cleaning is necessary, especially if the fluid is hot.
2. A sump strainer does not have an indicator to tell when it is dirty or clogged.
3. A sump strainer may block fluid flow and starve the pump if not sized correctly and properly maintained.
4. A sump strainer does not protect components downstream from the particles generated by the pump and it only prevents ingestion of gross particule sizes.

SUCTION FILTER

A suction filter is placed in the pump's inlet line outside or above the reservoir. Range of filtration commonly used in suction line filters is from 238-25 micrometres. (Figure 10-6)



(Figure 10-6)

ADVANTAGES:

1. a suction filter protects the pump from dirt in the reservoir.
2. indicators signaling the need for service are generally provided.
3. the filter element can be serviced without disassembling the suction line or reservoir.

DISADVANTAGES:

1. a suction filter may starve the pump if not sized or

engineered into the system properly.

2. a suction filter does not protect components downstream from the particles generated by the pump.

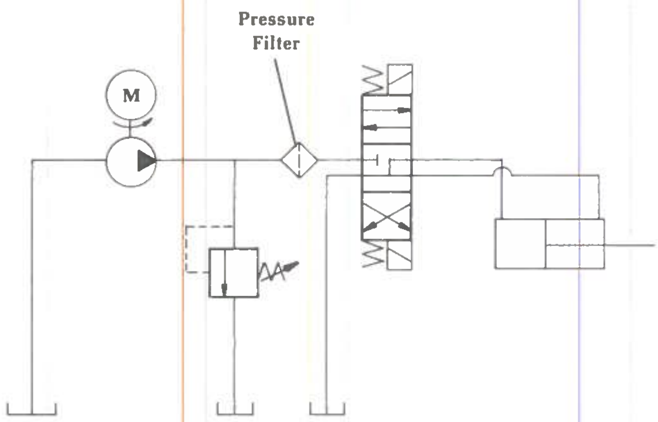
3. filtration limited by maximum available vacuum at pump.

PRESSURE FILTER

A pressure filter is positioned in the circuit on the outlet side of the pump. (Figure 10-7)

Range of filtration usually found in pressure line filters is from 40 micron to sub-micronic.

A pressure filter can also be positioned between systems components. If the flow between the components can flow in two directions (as between directional valve and cylinder), the filter must be capable of handling bi-directional flow. Bi-directional pressure filters are used on the downstream side of servo valves and in closed looped hydrostatic transmission.



(Figure 10-7)

ADVANTAGES:

1. a physically small pressure filter can filter very fine particles since system pressure is available to push the fluid through the element.

2. a pressure filter can protect a specific component from the harm of deteriorating particles generated from upstream components.

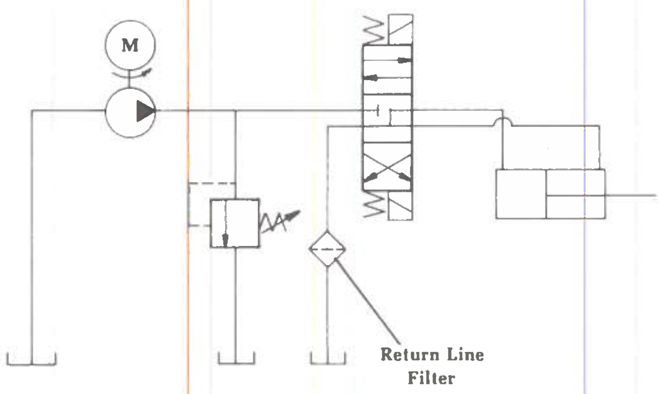
DISADVANTAGES:

1. the housing of a pressure filter must be designed for high pressure because it is operating at full system pressure. This tends to make the filter expensive.

2. if pressure differential and fluid velocity are high enough, dirt can be pushed through the element or the element may tear or collapse.

RETURN LINE FILTER

A return line filter is positioned in the circuit just before the reservoir. Range of filtration usually found in return line filters is from 40-3 micrometres. (Figure 10-8)



(Figure 10-8)

ADVANTAGES:

1. A return line filter catches the dirt in the system before it enters the reservoir.

2. The filter housing does not operate under full system pressure and is therefore less expensive than a pressure filter.

3. Fluid can be filtered fine since system pressure is available to push the fluid through the element.

DISADVANTAGES:

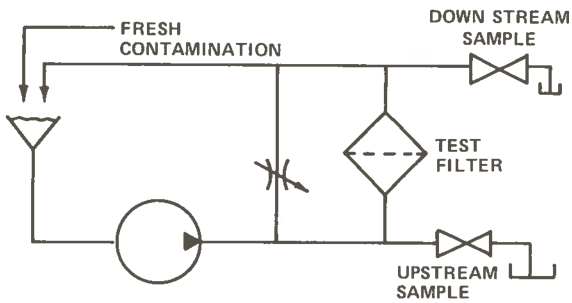
1. There is no direct protection for circuit components.

2. In return line full flow filters, flow surges from discharging cylinders, actuators, and accumulators must be considered when sizing.

FILTER TESTING AND RATING

Until recently, no reliable method of filter testing was available. However, with the program instituted by manufacturers and users of hydraulic filters at the Basic Fluid Power Research Center, (BFPR) at Oklahoma State University, a reliable method has been developed. The BFPR has developed a modified multipass test for evaluating the filter performance of a fine hydraulic filter element. The test is formalized as the National Fluid Power Assoc. (NFPA) standard T3.10.8.8.1973. It is also recognized by the American National Standard Institute (ANSI) number B-93.31-1973.

The multipass filter test determines the "efficiency" of a filter. A specific test stand was developed to eliminate many variables in the test. A schematic of the test circuit is shown. (Figure 10-9)



(Figure 10-9)

Referring to (Figure 10-9), the following conditions are maintained:

a. A closed loop system which causes any contaminant which escapes the test filter to be recirculated.

b. Continuous injection of fresh contaminants at a rate of 10 mg. per liter of rated flow.

c. Flow rate through the filter equal to its rated flow with system volume equal to 4 times the filter flow rate.

d. Provisions for taking upstream and downstream fluid samples.

Tests must be performed on a qualified test stand in which variables have been tested and proven to have eliminated by proper design.

The rating of a filter may be expressed in terms of a Beta number. This number is the ratio of upstream particles to downstream particles of a particular size or larger. In other words, it is an expression of the particle trapping effectiveness of a filter for that particle size.

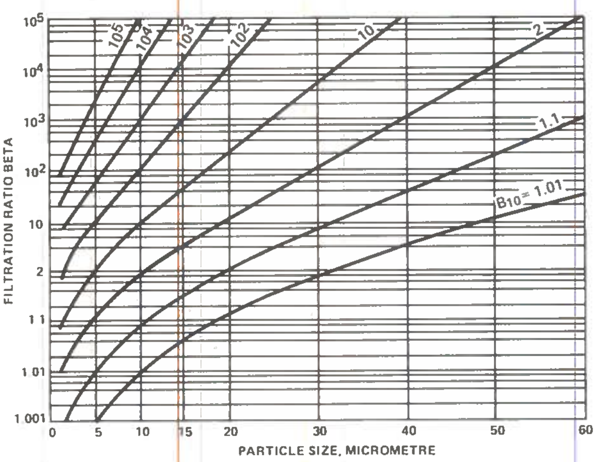
$$\text{Beta ratio} = \text{Bu} = \frac{\text{Number of particles upstream of u size or greater}}{\text{Number of particles downstream of u size or greater}}$$

PARTICLE DISTRIBUTION ANALYSIS
(AVG. NUMBER/ML. GREATER THAN INDICATED SIZE IN MICROMETRES)

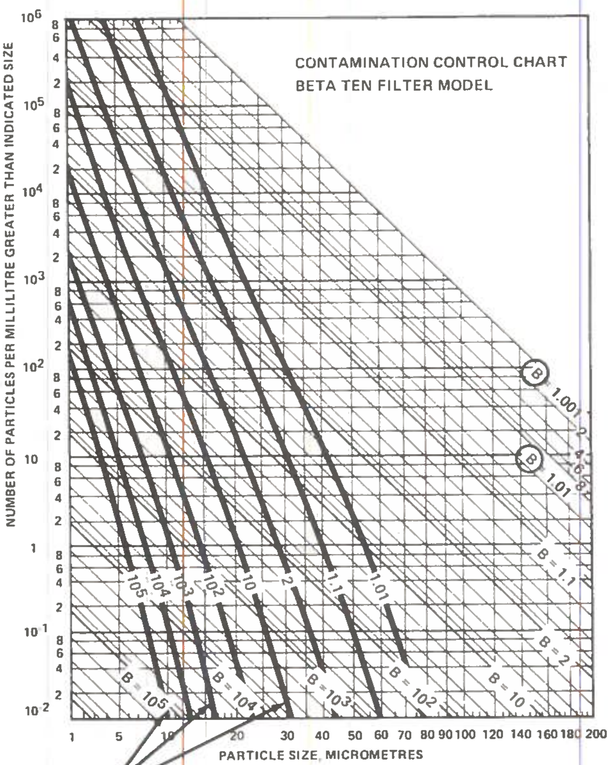
Sample		β 10	β 20	β 30	β 40	Beta 10
10%	Up	1738.6	225.8	56.84	19.04	25.32
	Down	68.7	8.3	1.52	0.46	
20%	Up	1677.9	234.8	60.44	22.08	86.85
	Down	19.3	1.0	0.22	0.06	
40%	Up	1717.4	232.5	67.56	24.56	73.96
	Down	23.2	1.31	0.24	0.10	
80%	Up	1798.6	249.4	68.92	24.68	30.15
	Down	59.7	7.8	2.20	0.88	

Minimum Beta 10	25.32	ACFTD Capacity	24.3 GM	Final Level In Reservoir	45.0 MG/L

(Figure 10-10)



(Figure 10-11)



These lines represent the theoretical performance of ideal B10 filters of various valves.

(Figure 10-12)

In determining filter efficiency, various valves of Bu are calculated. These values are B_{10}, B_{20}, B_{30}, and B_{40} values.

They are calculated at four different points in time throughout the life of the filter. These four points are after 10%, 20%, 40% and 80% of the maximum allowable pressure drop as specified by the filter manufacturer. The averages of the B values for the entire test may then be plotted on the same contamination control chart that was used to represent the particle size distribution of dirt in the system. (Figure 10-10) shows a typical sample of filter test data. This is an example of the Beta as tested and reported, showing the various test points.

From (Figure 10-10), the minimum Beta 10 is 25.32. However, Beta values are also important at larger particle sizes. In general, if we were to plot particle size verses filtration level for varies Beta 10 values, it can be seen that the filtation value would seem to increase. Consider the following graph. (Figure 10-11)

Enter the graph at the bottom at 10 micrometre particle size. Moving upwards we encounter lines running at a 30° angle.

These lines are a graphical representative of $Beta_{10}$ filters at various efficiencies. Moving up the 10 particles size line to B_{10} = 2 will allow us to find the filter's affectiveness on 10 micron particles. We find the value of filtration ratio = 2. Now, instead of moving up the 10 particle size line, move up to the 20 particle size to the B_{10} =2 line. Moving to the left from the intersection point to the filtration ratio, we find the filtration ratio equal to 10. This value should be read as B_{20} = 10. What we have found here is that this particular filter with a $Beta_{10}$ of 2, will have a $Beta_{20}$ of 10. In other words, this filter will effectively trap 9 out of every 10 particles of 20 micron size.

The need for obtaining such values at other than the B_{10} point is due to the fact that filters may perform better or worse at other particle sizes. This may be due to the presence of poor seals, imperfections in the filter media or a leaking bypass valve. Because of this, all points should be checked and compared with ideal filter performance data.

Another chart which displays very much the same data as (Figure 10-11) is (Figure 10-12). However, here we have added the number of particles along the vertical axis. Also found is particle size on the horizontal lines of constant, Beta at a 45° angle, and ideal B_{10} lines of performance at a 60° angle. This curve relates how a $Beta_{10}$ filter of a particular value will perform with other than 10 particles.

EXAMPLE: (10-1)

Filter Company X-Y-Z rated their filter at $Beta_{10} = 10$. What will be the theoretical rating of the filter at B_5, B_{20}, B_{30}, and B_{40}?

SOLUTION:

From (Figure 10-12), we may find the solution. Entering at the $Beta_{10} = 10$, we draw lines where it intersects the various particle sizes (see 10-13). From the intersections, we find:

$B_5 = 2.2$
$B_{20} = 250$
$B_{30} = 6000$
B_{40} = off the graph

The values obtained from the chart were for an ideal filter. These values should be compared with experimental data, to see how the real filter compares to ideal performance. Of course, the closer the correlation, the better the filter.

FILTER BYPASS

If a filter maintenance is not performed, the element will tend to load up with contaminant and the pressure differential across a filter element will increase.

A larger increase in pressure differential across a filter on the suction side of a system means that a pump will eventually cavitate.

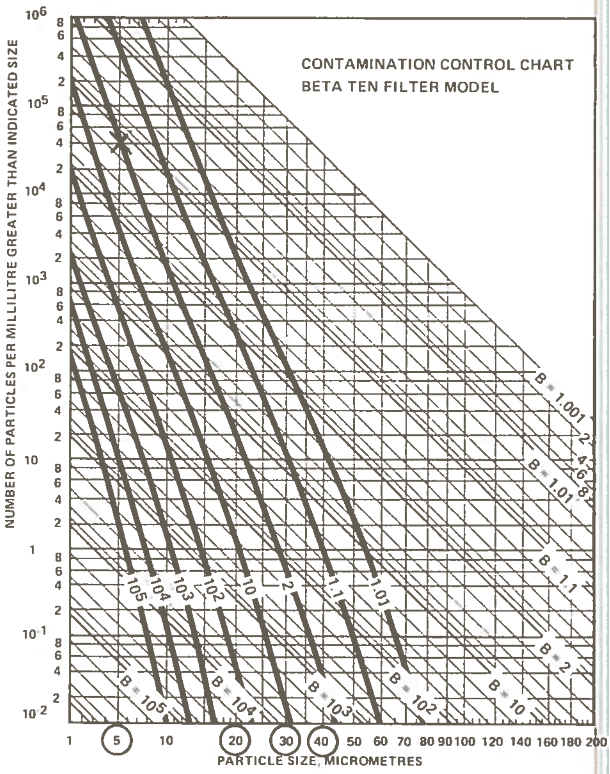
A large increase in pressure differential across a filter on the pressure side of a system means that the filter element will eventually collapse.

To avoid these situations, a simple or direct acting relief valve is used to limit the pressure differential across many filters. This type relief valve is generally called a bypass valve.

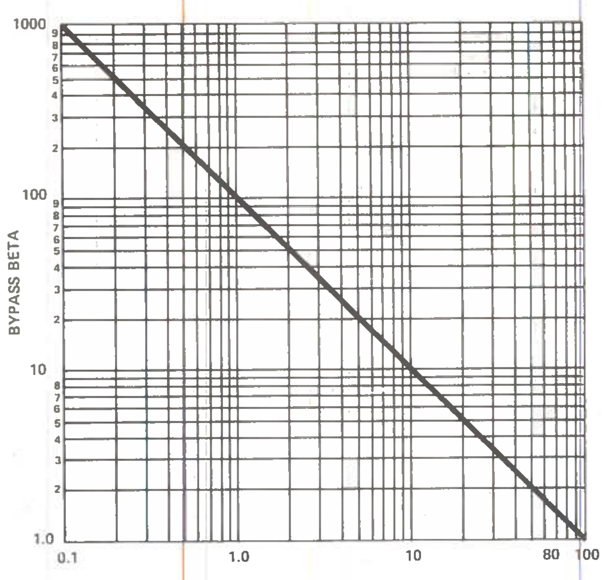
When the maximum safe differential pressure is reached across the filter, the bypass valve will begin to open, thus directing part of the system flow around the filter, rather than through it. This prevents catastrophic element failure but results in a very adverse effect on filter performance.

INFLUENCE OF BYPASS ON FILTER PERFORMANCE

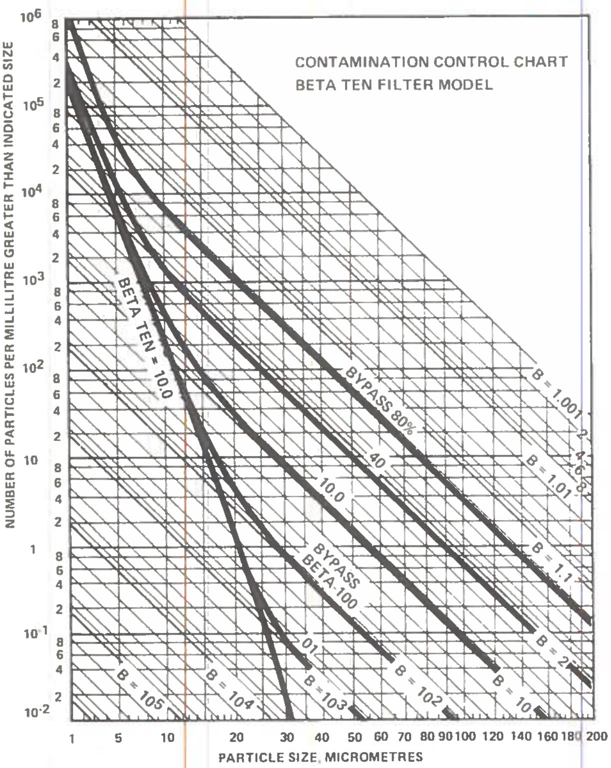
The effective filtering of a fluid is based on the fact that the flow entering the filter is passing through the filter media. As partial flow is bypassed, either through leaky seals or directly through a bypass valve, the filtration performance is decreased.



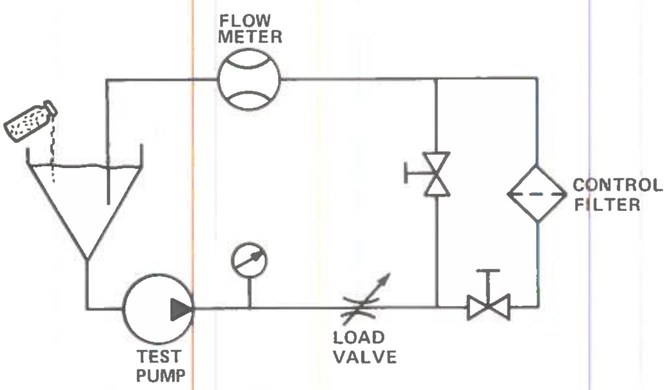
(Figure 10-13)



(Figure 10-14)



(Figure 10-15)



(Figure 10-16)

The amount of decrease is related to percent of bypass. This is shown in (Figure 10-14). The horizontal axis is the % bypass and the vertical axis is the bypass Beta referring to the 45° lines on (Figure 10-14). The bypass Beta at various values is shown.

% Bypass	B_{10} Value
a. .1% =	1000
b. 1% =	100
c. 10% =	10
d. 80% =	1.3

If we now plot these valves on (Figure 10-13) for a $Beta_{10}$ filter, we may see how it degrades as it begins to bypass (Figure 10-15). By employing (Figure 10-15) the significance of system bypass can be easily evaluated.

CONTAMINANT SENSITIVITY

The reasons why a filter is installed in a hydraulic system are to extend component life, enhance the life of the oil, and reduce maintenance dowtime. What new advances are being made in analyzing a component's sensitivity to various size contaminants in the fluid power system?

Major hydraulic component manufacturers throughout the United States are beginning to run contaminant sensitivity tests on their components. Using controlled contaminant ingression rates and specified pressure and flow parameters, a tolerance profile of the component is generated.

The component's contaminant sensitivity is a measure of the component's life when subjected to varying sizes of particulate matter. It is expressed as a profile for a specific degradation of flow performance.

Of the components already discussed in this text, one of the more susceptable ones is the hydraulic pump. By a procedure outlined by the ANSI Standard, (Figure 10-16), a pump may be subjected to a pump sensitivity test whereby its contaminant sensitivity can be found. During the test, various particle sizes (0-5, 0-10, 0-30 micrometres, etc.) are injected at a rate of 300 mg per litre of the rated flow. After each contaminant exposure, the fluid is "cleaned". Fluid is cleaned by letting it pass through highly efficient filters removing particles greater than 5 micrometres. Particle distribution is then checked with a particle counter to insure its cleanliness. This is done to eliminate the variable of remaining dirt causing additional degradation of flow. Flow rate is then measured after each exposure and compared with the original flow. A ratio (Qf/QR) of degradation is calculated. When the results

obtained in this test are plotted on a suitable chart, (Figure 10-17) will result.

CONTAMINANT SENSITIVITY

This graphically illustrates how a pump's performance will degrade when different particle sizes of AC fine test dust are injected into the fluid stream. In this way, pumps may be rated in terms of % survivability. To better understand (Figure 10-18) let us look at an example. NOTE: survivability means that N% of the pumps tested were less susceptable to the test at tested operating conditions.

EXAMPLE: (10-2)

A manufacturer is selling a gear pump and states that it has a survivability of 40%. How much did this pump degrade as the following size AC test dust was injected as per the pump test.

a. 0-10
b. 0-20
c. 0-30
d. 0-40
e. 0-80

SOLUTION:

The 40% pump is designated by the 40 at the angled lines. The particle range at the verticle left and the degradation flow ratio at the right. By reading directly we find:

a. 0-10 No degradation
b. 0-20 99.9%
c. 0-30 99%
d. 0-40 97%
e. 0-80 80%

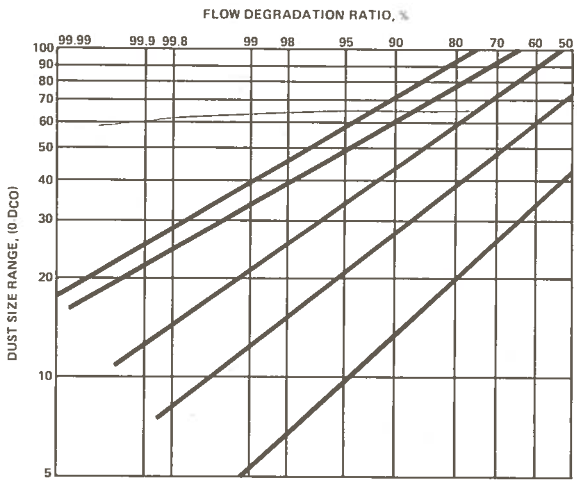
HOW TO SELECT THE CORRECT BETA$_{10}$ FILTER

When selecting the correct filter for a system, there are three parameters which we must consider:

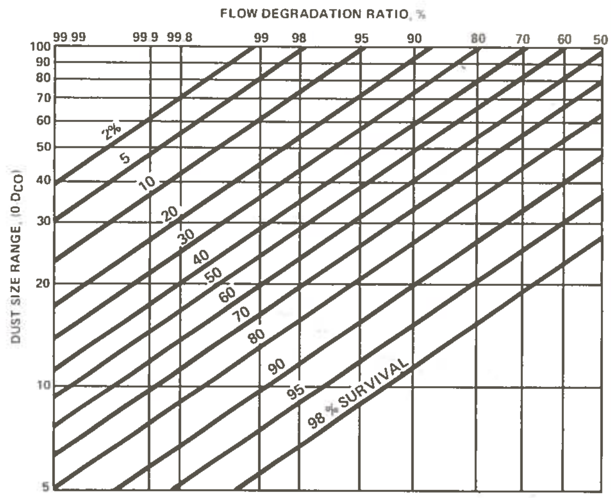
1. The survivability of the particular pump.
2. The ingression rate per minute greater than 10 micrometres entering the pump.
3. The pressure at which the pump operates.

The survivability of the pump must be known. This will be given in terms of survivability. This terminology has already been discussed.

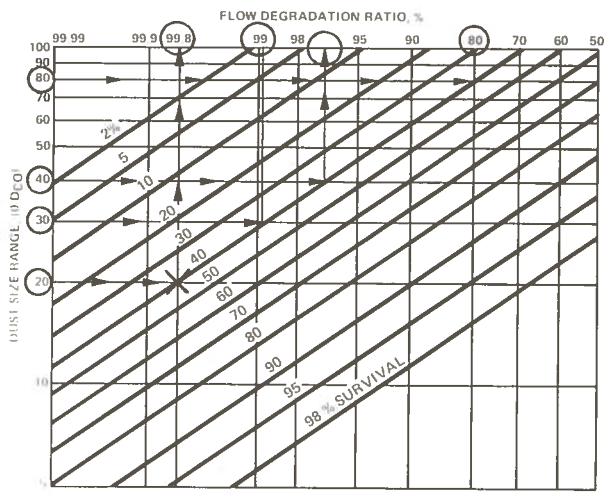
The ingression rate of particulate matter into the pump assumes 10^8 particles per minute greater than 10 micrometres. For the industrial system, we will assume that it will not exceed 10^8 value. If, however, a very dirty environment is encountered for a



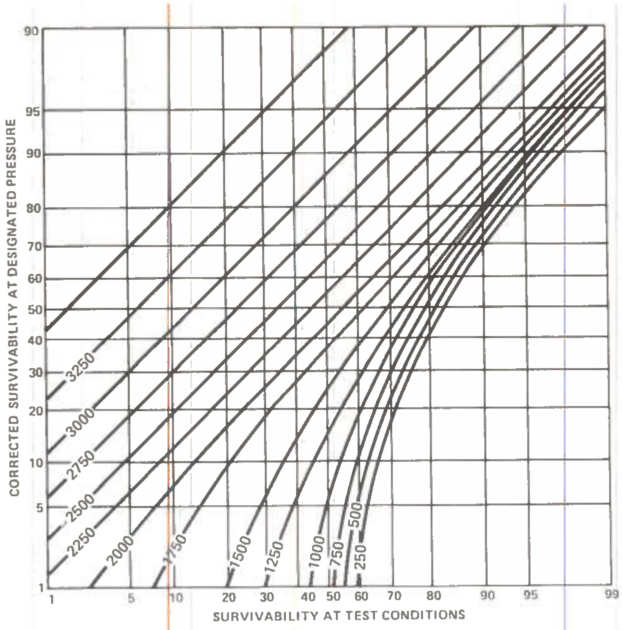
(Figure 10-17)



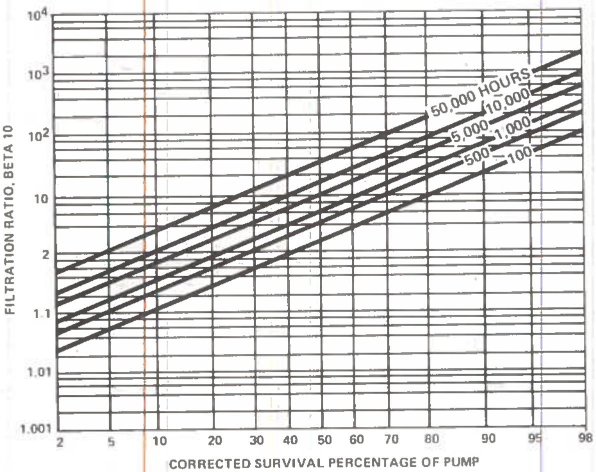
(Figure 10-18)



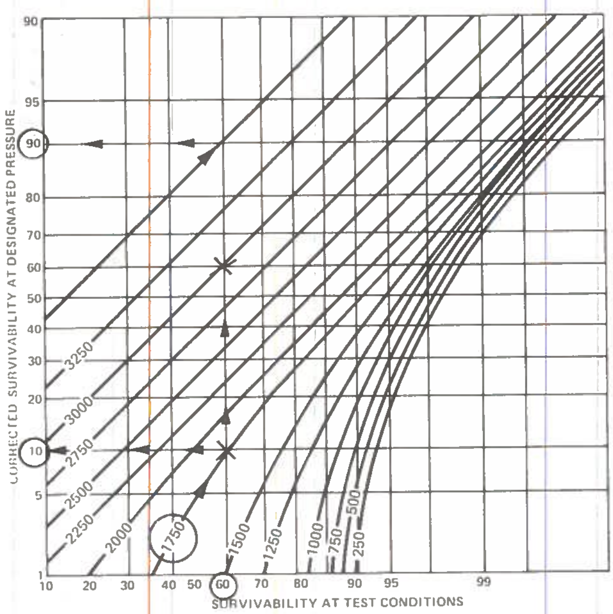
(Figure 10-19)



(Figure 10-20)



(Figure 10-21)



(Figure 10-22)

particular machine, the life of the components will have to be derated from the calculated value.

The pressure at which a pump is run is a significant factor. In order to relate how pressure will effect the life of a pump, consider (Figure 10-20).

(Figure 10-20) shows graphically how variations in pressure will affect survivability. Along the horizontal axis, we have survivability at test conditions (test conditions are related to the manufacturer's testing). This axis may be moved left or right in order that test condition pressure and survivability coincide.

For example, if we had a pump that had a 50% test survivability run at 2250 psi and operated at 1500 psi, its survivability would increase to 25%. If we took the same pump and operated it at 2500 psi, its designated survivability would be derated to 60%.

With the corrected survivability, we can now enter a graph to determine the filter Beta rating for a given number of hours with a rated pump, (Figure 10-21). With this figure, a life (in hours) is selected, intersected vertically with the corrected survival percentage, and the $Beta_{10}$ rating is read at the left.

EXAMPLE: (10-3)

The survivability of a particular 20 gpm pump is 60% when run at 3000 psi. It is intended to be used in an industrial application and the running life must be at least 10,000 hours. What $Beta_{10}$ rating would you suggest if one pump is run at 1750 psi and the other at 3500 psi?

SOLUTION:

First, an assumption that the industrial system will ingress less than 10^8 particles per liter greater than 10 microns is made. We may now find the corrected survivability of this pump at the two pressures. (Figure 10-22)

Entering the curve at 60% survivability at test conditions, we first manipulate the curve to correspond to the pump. The vertical lines (horizontal axis) is moved to the left so that 60% and 3000 psi coinciding. This corrects the plot for test conditions.

Now you travel vertically upwards, until we intersect the desired pressures of 1750 and 3500 psi. We then move horizontally to the left and find the corrected survivability to be 10% and 90% with these values we may use (Figure 10-23). Enter at 10% and 90% and move vertically upward until the 10,000 hour line is intersected. Moving to the left, we find that this pump, if run at 1750 psi, will last 10,000 hours with a $Beta_{10}$ of 2. The same pump operating at 3500 psi

with a life expectancy of 10,000 hours will take a filter with a $Beta_{10}$ rating of 200. It is evident that there is a measurable difference due to the operating pressure.

THE FILTER IN THE SYSTEM

We now know the Beta rating of the filter needed in our system, but still do not know its particular size. The size is correlated directly to the flow rate of the fluid and areas needed to be sure that a particular pressure drop is not exceeded.

NECESSARY SIZING INFORMATION

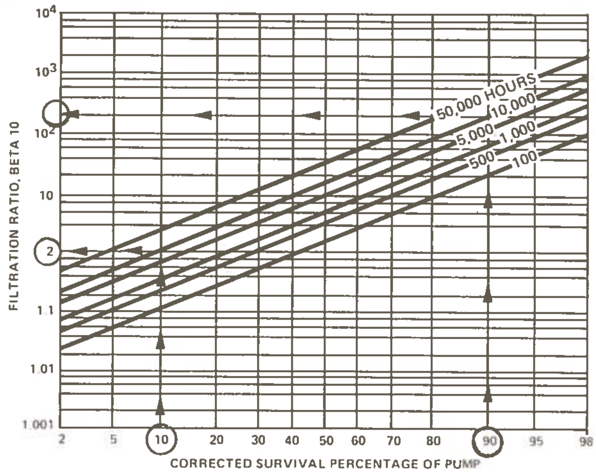
Important data which must be considered is the following:

1. FLUID — the fluid must be defined. The viscosity of the fluid, its chemical nature, and specific gravity are needed. The Parker Hannifin Design Engineer's Handbook is a compilation of fluids with their specific gravity and viscosity at two points.
2. TEMPERATURE — determine the minimum and maximum operating temperature ranges for the filter environment.
3. FLOW — for suction and pressure filters (or filters used at points in the circuit with constant flow rates) determine the pump maximum rated flow. For return line filters, (or filters used at points in a circuit variable flow rates) also check the system and its operation of components such as large hydraulic cylinders or accumulators, that, when discharging, can significantly affect the flow rate through the filter.
4. PRESSURE — determine the complete pressure environment of the filter. Pressure peaks or shocks in the system should not be noted.

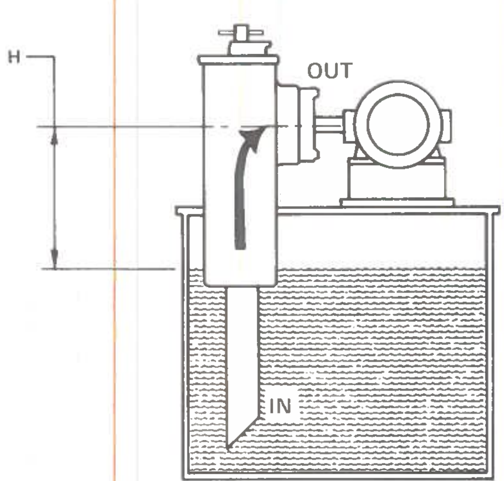
SIZING OF SUCTION FILTERS

When sizing a suction line filter, we must consider the flow rate through the filter, the lift of the pump, the pressure drop caused by the inlet line, and the pressure drop caused by the filter. In suction applications, should the pressure drop through the filter be too high, there is a chance that the pump may cavitate, if a bypass is not correctly sized. The determining factors in a conventional arrangement are:

1. lift
2. pressure drop through filter
3. pressure drop through piping
4. fluid characteristics & temperature



(Figure 10-23)



(Figure 10-24)

LIFT

Lift is the amount of energy needed to raise the fluid to the level of the pump. To calculate the lift restriction, the following equation can be used:

LIFT (in.Hg.) = 0.733 x height (inches) x specific gravity

L (in.Hg.) = 0.733 x H x SG
(10-1)

When considering the height H, it is the distance from the surface of the oil to the inlet port of the pump. This is shown in (Figure 10-24).

The specific gravity of the fluid may be found in the Design Engineer's Handbook, which is a compilation of hydraulic fluids, with their specific gravities and viscosity at two points.

MAXIMUM VISCOSITY OF THE OIL

The first parameter that must be determined is the maximum viscosity of the oil. This is needed so that one may determine the maximum pressure drop through conductors and filtrate maximum viscosity. This is obtained by 1) noting the oil, 2) its minimum temperature, 3) consulting the fluids charts, 4) plotting the values on the ASTM Standard Viscosity — Temperature Charts. Let us work an example on the above procedure.

EXAMPLE: (10-4)

Mobil DTE 24 is being used in one of our systems. Its minimum temperature is 60° F. What is its maximum viscosity?

SOLUTION:

By consulting the fluids chart in the Design Engineers's Handbook, we find Mobil DTE 24. Its specific gravity (For lift calculation) is .871, with a viscosity of 153 and 44 at 100° F and 210° F respectively. We plot the two values of viscosity on our ASTM Standard Viscosity Paper. (Figure 10-25). By drawing a line through the two given viscosities at their particular temperature, the viscosity index of the oil is found.

Then by intersecting the minimum temperature with the viscosity index and moving to the left, the maximum viscosity of 500 SUS is obtained.

This maximum viscosity is used to calculate the maximum pressure drop and make sure that it does not exceed the maximum allowable vacuum specification for the pump. Also, pumps have a maximum viscosity specification. For gear, vane, and piston, it is typically 7000 SUS, 4000 SUS and 1000 SUS respectively. This must be adhered to so that

3. A 3 in.Hg. or a 4.5 in.Hg. spring for a life of 2 or 3 can be selected.
4. If $1\frac{1}{2}$ in. or greater diameter pipe is selected, the pressure drop through the pipe is less than .25 in.Hg.

SIZING PRESSURE FILTERS

Pressure filters are designed to handle the full rated pressure of the filter.

Fluid velocities in the piping should be kept below 15 feet per second to aid in heat dissipation. The pressure filter should be located just before those components it is designed to protect, but should be positioned, wherever possible, to avoid surges and shocks from such components as quick acting relief valves or accumulators (i.e. in front of the relief valve or accumulator, instead of after it. Rapid changes in flow or differential pressure may force some contaminant through the filter media.

The clean flow pressure drop should be below 5 psi, where possible, and the dirty pressure drop kept less than 25 psi to allow for more efficient use of the element. The consequences of installing a bypass relief valve should be considered in a pressure line application. If no bypass is used, the differential pressure will continue to rise as dirt is added. At some point, pressure drop across the filter will increase. The flow through the unit may decrease, being directed through a relief valve or other elements of the system. The filter element will collapse. When a filter element collapses, large amounts of dirt located on the filter element will be sent downstream, which can cause wear or catastrophic component failure. Installing a suitable bypass around the filter, or using a filter with a built-in bypass will prevent this. However, the bypass may also open when the flow rate increases or if the viscosity rises.

Where a filter is used with no bypass design, it is good practice to use an electrical dirt indicator to provide an emphatic signal of the element's condition.

The life of the filter element is proportional to the amount of filter element area.

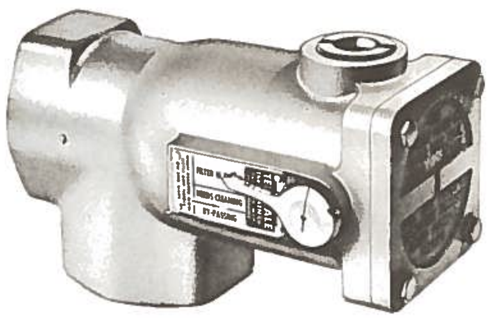
Where possible, the filter life factor should be at least 4 to 5 (life being the ratio of dirty pressure drop to clean pressure drop) for pressure filter.

SIZING RETURN LINE FILTERS

Return line filters must be sized to work properly. Just because the pump is 20 gpm, doesn't mean the



(Figure 10-29)



(Figure 10-30)



(Figure 10-31)

return line filter should be sized at 20 gpm. Due to discharging components within the system, flow rates through the return line can vary by 500%. A cylinder with a 2:1 rod can discharge at twice the flow or more, depending on the time cycle as it is changed. Having several cylinders operating at the same time, will increase the return line flows. If accumulators are dumped through the return line, they too will increase flow rates.

To size a return line filter, calculate the maximum discharge flow rate and use that flow rate in selecting the proper housing. Fluid velocities in return lines should be kept below 10 feet per second to limit velocity into the filter. The clean flow pressure drop should be below 5 psi, where possible, and the dirty pressure drop should be below 25 psi to give good filter performance.

If the return line filter is the last component in the system, then the filter housing must be designated to handle at least the differential pressure across the filter. If another component, such as a heat exchanger is downstream of the return line, then the filter housing must be strong enough to contain the accumulated pressure drops of the filter element and that of the heat exchanger. Since the filter, and any other components, add restriction to the return line which in turn causes backpressures, this added back pressure must be considered in the response of other system components which may have back pressure in sensitivities.

CONCLUSION

Filtration is probably one of the most important components added to a fluid power system. Types of filtration are:

1. suction
2. pressure
3. return time

When selecting a filter, items should be considered:

1. the survivability of the pump
2. the operating pressure
3. the expected life
4. the type of environment
5. the type of filtration (suction, etc.)

With this and the procedure outlined in Chapter 10, the correct filter can be selected.

cavitation may be avoided. Also operating viscosity may be checked. It should be typically around 100-130 SUS at operating temperature.

MAXIMUM ALLOWABLE VACUUM

This value should be determined as outlined in Chapter 4. After this is obtained, we must allot a particular amount of this available pressure for the filter and the necessary filter piping. It is good engineering practice to allot .25 in.Hg. for piping and the rest for the filter.

The remaining available pressure is to be considered the pressure drop associated with a dirty filter. The life factor for a suction filter should be between 2 and 3 times the pressure drop of a clean filter. Therefore, by dividing the remaining pressure by 2 to 3 we will have the clean pressure drop for a particular element. Catalog data should be consulted for the exact filter needed. By following this procedure, the chances for cavitation can be minimized.

FLOODED SUCTION

Since flow restrictions are very critical in suction filters, reducing the effect of any will extend filter life and decrease the possibility of cavitation. Raising the liquid level to a point above the level of the pump inlet can increase the pressure available for the filter element and decrease the risk of cavitation.

The calculation of the total restriction is the same with one exception of lift. When the liquid level is above the pump inlet, this positive liquid level exists. This positive liquid level, called "head", works as an advantage, but is calculated in the same manner as lift.

DESIGN OF THE SUCTION FILTER

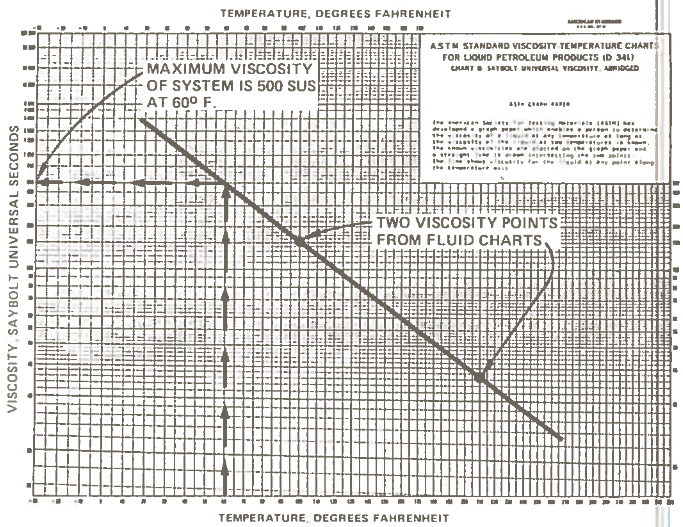
Here we will go through the procedure that need be followed when designing for a filter.

EXAMPLE: (10-5)

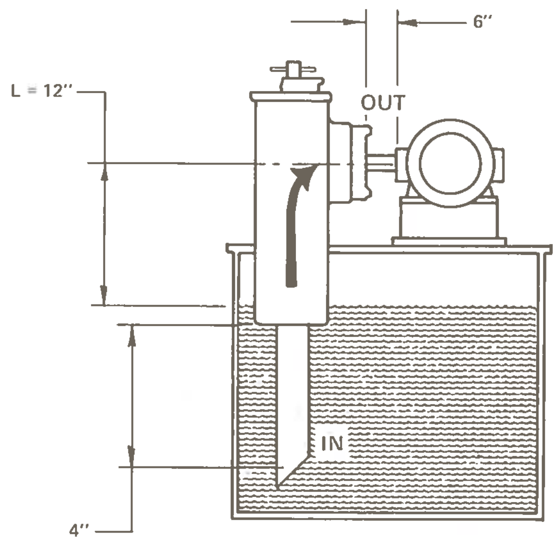
A 20 gpm pump with a survivability of 50% under test conditons of 2150 psi, is operated at 1000 PSI. Its service life at rated pressure and RPM is to be 10,000 hours. It will be fitted in the following system (Figure 10-26). The oil is DTE 24 with a minimum deisgn temperature of 60° F. The location is 1000 ft. above sea level and the maximum allowable vacuum of the pump is 7 in.Hg. DESIGN THE SYSTEM.

SOLUTION:

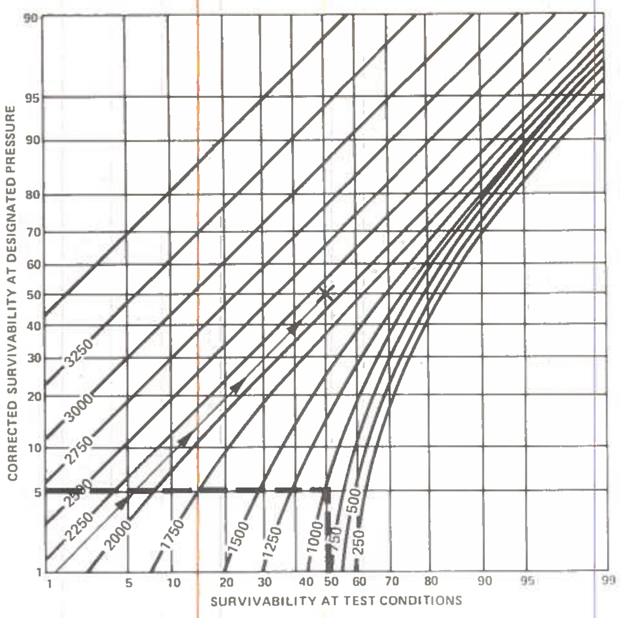
First, we will find the $Beta_{10}$ rating of the filter that



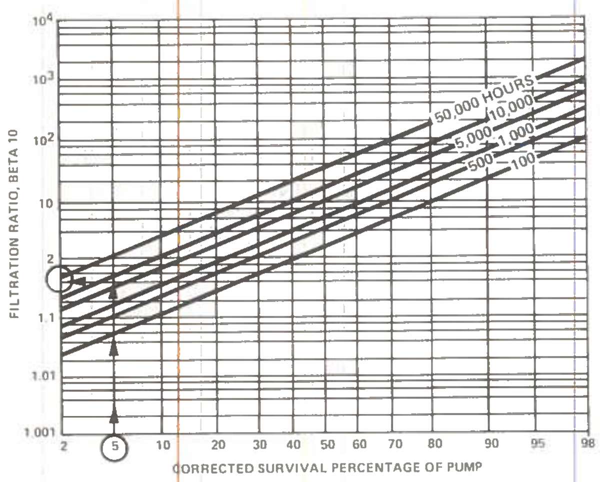
(Figure 10-25)



(Figure 10-26)



(Figure 10-27)



(Figure 10-28)

will deliver the stated life. This means that we must find the corrected survivability from (Figure 10-27). From this figure we find the corrected survivability to be 5%.

Entering (Figure 10-28) we find the $Beta_{10}$ rating equal to 1.6. This means that the filter must have a Beta of at least 1.6 to give the expected life. In this case, it happens to be a 20 C paper/depth element.

Next, we find the maximum available pressure drop for the associated filter parts. From Chapter 4, equation (4-3) we find the N.P.S.H.

N.P.S.H. = 30 - 7 = 23 in.Hg.

Maximum vacuum gage reading is found from equation (4-4).

MAXIMUM VACUUM GAGE READING = Atmospheric pressure at site - NPSH

Maximum gage reading = 28.8 - 23 = 5.88 in.Hg.

This value 5.88 in.Hg. is the maximum pressure drop that may exist in the piping, the dirty filter, and the necessary lift, without causing a cavitation problem.

The lift is found using equation (10-1) (5g given)

L(in.Hg.) = .0733 x H x Sg

L(in.Hg.) = .0733 x 12 x .871 = .766 in.Hg.

Subtracting .766 from 5.88 we have 5.12 in.Hg.

This means that for pressure drop due to pipe and dirty filter, we have available 5.12 in.Hg.

Subtracting .25 in.Hg. for line losses, we have 4.87 in.Hg. for filter. Dividing 4.87 by 3 for a superior life factor, we find the filter pressure drop equal to:

FILTER PRESSURE DROP = 4.87/3 = 1.61 in.Hg.

This means that filter with a 1.5 in.Hg. clean pressure drop will work effectively and exhibit excellent life. The bypass spring should be sized for 3 in.Hg. for a life factor of 2 or 4.5 in.Hg. for a life factor of 3.

We should now check to see that no more than .25 in.Hg. is used in the piping. We have a total of 10 in. of pipe. From pressure drop charts at 500 SUS with 1½ in. pipe or greater, the pressure drop will be less than .25 in.Hg.

Summarizing, we found that:

1. In order to obtain necessary life, we had to use a filter with a $Beta_{10}$ rating equal to 1.6.
2. The filter must have a maximum pressure drop when dirty of 4.5 in.Hg. with 500 SUS at startup for a life of 3.

CHAPTER 11

FLUID CONNECTORS

Relief valves, flow controls, pumps, motors, cylinders and other fluid power components have little reason for existing for their own sakes. They are designed to be used together in a fluid power system.

Many times, the method of connecting interacting components of a system is a minor consideration. We know that there must be some sort of fluid-carrying link between components, but the way it is done is sometimes considered arbitrary. This attitude may result in a system which is inefficient, unsafe, expensive, and difficult to service.

Conductors of a fluid power system are basically of two types: flexible and rigid. In this lesson we will consider each type.

In the analysis of the fluid power system, we are most interested in the pressure drop caused by fluid flowing through these conductors. To make it easier to compute the pressure drop in the system, tables have been compiled. A cursory explanation of the tables follow. For a more complete understanding of the tables, read the section on "TABLE DERIVATION."

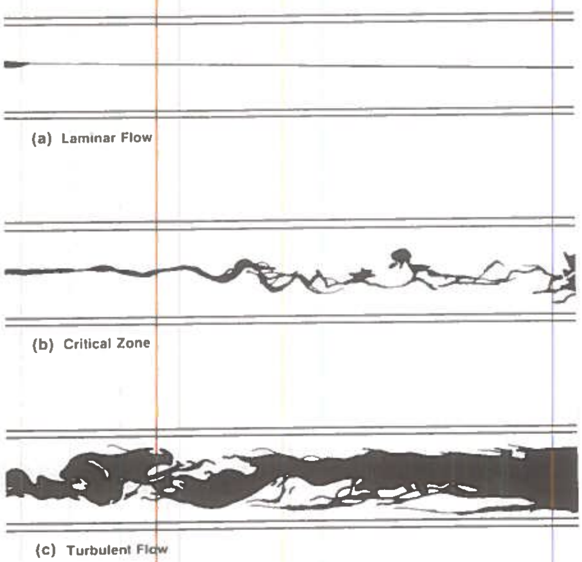
EXPLANATION OF FLOW VS. PRESSURE DIFFERENTIAL TABLES

Pressure drop, Tables 1 through 9 found in Parker Design Engineer's Handbook, indicate estimated pressure differential per linear foot of pipe, tube and hose for fluid flows up to 200 GPM at ten viscosities. Values were determined by computer and are rounded off to the fourth decimal place.

Tables 1 through 6 are calculated in units of PSI. In tables 7 through 9, pressure units are indicated in terms of inches of mercury (in.Hg.): this is used for calculations at the pump suction side.

Fluid flow in a conductor can exist in either a laminar or turbulent form. It can also be in transition between laminar and turbulent. These conditions of flowing fluids are indicated on the table by a heavy line running through each chart from top to bottom. To the right of the line flow is laminar. To the left, flow is turbulent. Shaded areas are transitional regions known as critical flow zones.

In the illustration, a reproduction of flow in a circular plastic-pipe is shown. A dye is injected at a point midstream into the flow. A series of drawings illustrate laminar flow (a), the critical zone, (b) and finally turbulent flow (c) as fluid velocity progressively increases.



(Figure 11-1)

An increase in velocity is not the only cause affecting a change between laminar and turbulent. Factors such as pipe diameter, specify gravity (fluid density), and viscosity also affect laminar and turbulent states.

Reynolds number is a dimensionless number which takes into account the effects of velocity, viscosity, specific gravity, and pipe diameter on a flowing fluid. With a Reynolds number of less than 2000, flow is laminar. As Reynolds number increases, flow tends to become turbulent, with the critical zone existing between 2000 and 5000. Under normal conditions, flow will be fully turbulent as Reynolds number equals 5000.

Values given for the critical zone in the tables are for fully the specific flow conditions.

When turbulent flow is obtained, a rapid mixing takes place. Compared with the smooth flowing action of laminar flow, this action causes additional pressure energy to be used to maintain flow.

At times, this results in an apparent contradiction in the tables when going from turbulent flow or the critical zone, to laminar flow (left to right on the chart). For example, with fluid velocity remaining constant, pressure differential at 200 SUS might be more than at 300 SUS for a specific conductor size. This is because flow is turbulent at 200 SUS and laminar at 300 SUS.

CALCULATING PRESSURE DIFFERENTIAL IN STRAIGHT LINE LENGTHS

To calculate fluid flow pressure differential in straight line lengths lying in a horizontal plane, several factors must be known. These are conductor size and length, flow rate, fluid viscosity and specific gravity.

EXAMPLE: (11-1)

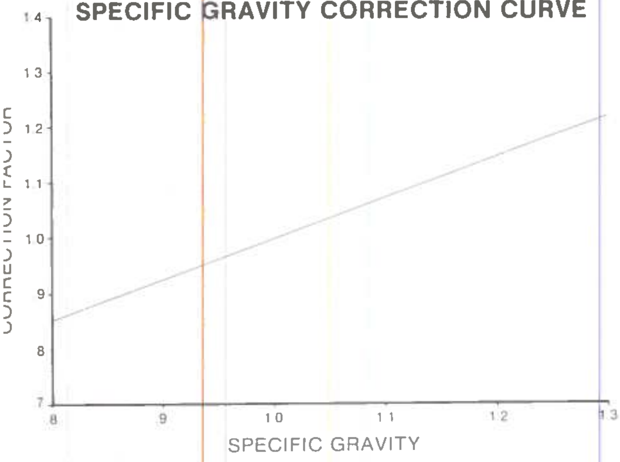
Assume that 50 GPM is flowing through 45 ft. of 1¾ in. tube with a .083 in. wall. Fluid specific gravity is .865. Find the pressure differential at a viscosity of 100 SUS and 400 SUS must be determined.

SOLUTION:

Referring to Table 5 in the Design Engineer's Handbook, we find a flow rate of 50 GPM through 1¾ in. tube with a .083 in. wall. The table indicates a pressure differential of .1282 PSI/ft. @ 100 SUS and .885 PSI/ft. @ 400 SUS. These values are based on the fluid having a specific gravity of 1.0. Since the fluid in our example has a specific gravity of .865, pressure differential in this instance will be less. To

Flow GPM	Tube				Velocity FPS	Viscosity — SUS	
	Size	OD	Wall	ID		100	400
50	1½	1 5	120	1 260	12 864	3800	4708
			134	1 232	13 455	4228	5151
			156	1 188	14 470	5026	5958
			188	1 124	16 165	6538	7435
	1¾	1 75	065	1 520	7 782	1152	1723
			072	1 606	7 918	1200	1784
			083	1 584	8 140	1282	1885
			095	1 560	8 392	1378	2004
			109	1 532	8 702	1502	2154
			120	1 510	8 957	1609	2283
			134	1 482	9 299	1758	2460
			156	1 438	9 876	2029	2775
			188	1 374	10 818	2519	3330

SPECIFIC GRAVITY CORRECTION CURVE



determine how much less, the Specific Gravity Correction Curve is referred to. The curve indicates that the correction factor for a .865 specific gravity is .9. This factor is then multiplied by the pressure differential values for 100 SUS and 400 SUS.

100 SUS P = .1282 PSI/ft x .9 = .1154 PSI/ft.

400 SUS P = .1885 PSI/ft x .9 = .1697 PSI/ft.

With 45 ft. of tubing, we need only multiply the PSI/ft value by 45 ft. to obtain the total pressure differential.

100 SUS P = .1154 PSI/ft. x 45 ft. = 5.2 PSI

400 SUS P = .1697 PSI/ft. x 45 ft. = 7.6 PSI

To determine pressure differential from the tables for horizontal straight line lengths, the following formula can be used:

Horizontal Straight Line Length P (PSI) = P(PSI/ft) @ Fluid Viscosity x Specific Gravity Correction Factor x Line Length (ft.)

(11-1)

CALCULATING PRESSURE DIFFERENTIAL IN FITTINGS & BENDS

As indicated, the tables show pressure differential per linear foot of pipe, tube and hose which are horizontal plane. They may also be used to determine the pressure differential through typical pipe fittings and 90° bends of tube and hose. This is accomplished with the use of a multiplier which transforms a fitting or bend into a length of straight conductor exhibiting an equivalent pressure differential.

Illustrated in the Design Engineers' Handbook is a chart which shows various pipe fittings and 90° bends with their appropriate multipliers. To determine pressure differential, the multiplier is used in the following formula:

Fitting or Bend P (PSI) = Fitting or Bend Multiplier x Conductor Inside Dia. (in.) x P (PSI/ft) @ Fluid Viscosity x Specific Gravity Correction Factor

EXAMPLE: (11-2)

Assume 50 GPM is flowing through 1¼ in. schedule 80 pipe. Viscosity is 100 SUS and specific gravity is .865. To calculate pressure differential for one elbow, the chart is referred to for the multiplier, which happens to be 2.5 From Table 2, 1¼ in. schedule 80 pipe has an inside diameter of 1.278 in. and a pressure differential per foot @ 50 GPM and 100 SUS of .3643 PSI. And, referring to the

correction curve for specific gravity, we find that .865 has a correction factor of .9.

To solve for P in one elbow, the formula becomes:

P $1\frac{1}{4}$ in.

Schedule 80 elbow (PSI) $= 2.5 \times 1.278$ in. x .3643 PSI/ft x .9

$= 1$ PSI

This indicates that a typical schedule 80 elbow will use approximately 1 PSI as 50 GPM passes through at 100 SUS. Now, assume that 50 GPM passes through $1\frac{3}{4}$ in. tube with a .083 in. wall at a viscosity of 100 SUS and a specific gravity of .865. Pressure differential for each 90° bend is calculated in the following manner:

P 90° Tube Bend (PSI) = 1 x .584 in. x .1282 PSI/ft x .9 = .2 PSI

Compared to the pipe elbow, this is a substantial pressure savings per bend.

TABLES DERIVATION

Internal flow in pipes exists in three different regimes: laminar, turbulent and a critical zone where flow changes from laminar to turbulent. (Figure 11-1) is a reproduction of flow in a circular plastic pipe with a streamer. This series of drawings show flow in its smooth laminar region (a) with its velocity gradually increasing and flow changing to turbulent and (b) finally turbulent flow. However, it is not only an increase in velocity that will cause flow to change from laminar to turbulent. Other parameters such as pipe diameter and fluid velocity will cause such phenomenon to take place. A term which takes into account all of the above three characteristics is defined as the Reynolds number.

$R = Vd/\nu$

Where V is velocity ft/sec
d is diameter in feet
ν is the kinematic viscosity in ft^2/sec

With the Reynolds number less than 2000, the flow will be laminar. As the Reynolds number is increased above this value the flow is apt to become turbulent, but not necessarily so. Between the values of 2000 and 5000, the critical zone exists. Unless extreme precautions are made, flow will move to fully turbulent near R = 5000.

When turbulent flow is obtained, a rapid chaotic mixing of the fluid takes place. More energy is lost (energy being directly related to pressure drop) due

to this violent mixing in turbulent flow, then is lost in smooth flowing action of laminar flow.

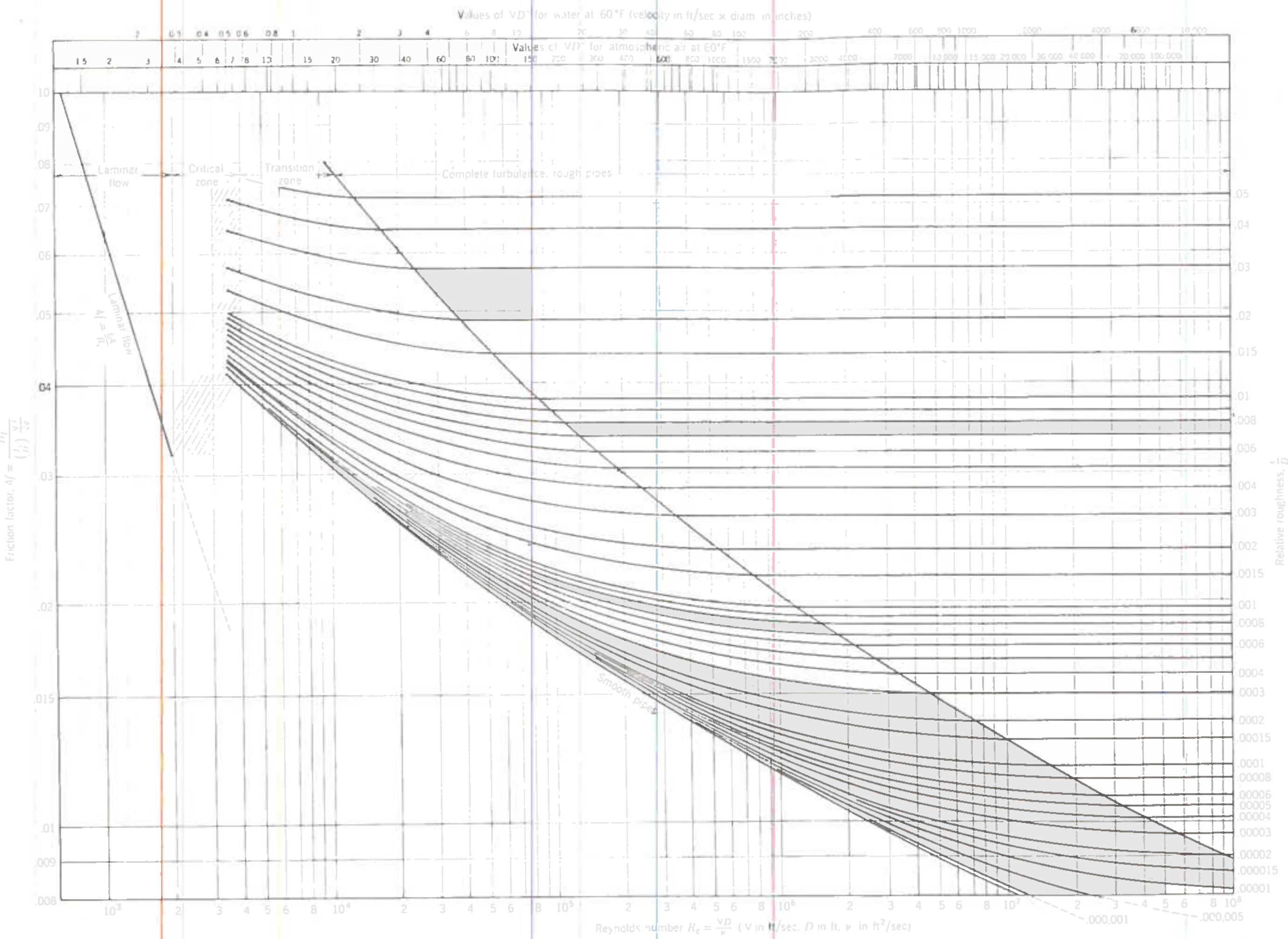
The pressure drop in pipe is characterized by the following equation:

$$P = .0808\ f\ l\ \frac{V^2}{d}$$

Where f is the friction factor
V is the velocity (ft/sec)
l is the pipe length (ft.)
d is the pipe inside diameter (in.)

The term V, the fluid velocity, can be easily obtained because this flow rate and pipe ID are known. However, in order to obtain the friction factor, a chart known as the Moody diagram must be consulted (Figure 11-2).

The Moody diagram is a chart which plots the friction factor of a pipe verses its Reynolds number for various roughness. Experimental results for pipes agree quite well with the Moody diagram.



(Figure 11-2)

From these charts many useful facts may be observed:

1. The straight line on the curve to the left represents the friction factor curve for laminar flow. The friction factor becomes:

$f = 64/R = 64\nu/Vd$

Inserting this into the pressure loss equation:

$$P = (.0808)\ (64)\ \frac{\nu}{Vd}\frac{l}{d}\ V^2 = 5.7\frac{\nu lV}{d^2}$$

From this above equation, it is apparent that as the viscosity and velocity V of the fluid increases, the pressure drop increases. As the diameter increases, the pressure drop decreases. It should be kept in mind that as long as the Reynolds number is less than 2000, the roughness of the pipe is inmaterial.

2. If we now turn to Reynolds numbers above 5000 and look at the lowest curve on the Moody diagram, we find it represents turbulent flow in smooth pipes. This is the curve which represents the characteristics of hose and tubing. For turbulent flow in smooth tube and hose, a simple friction factor equation can be derived from Blasius one-seventh power velocity equation which is:

$F = .3164/R^{1/4}$

This equation is frequently used in engineering calculations because of its simplicity.

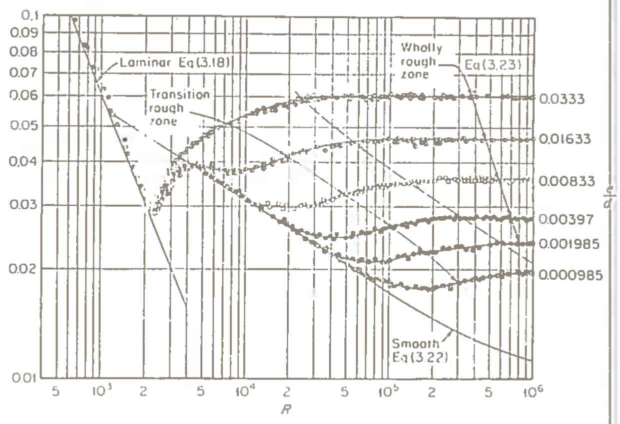
3. As we move from the smooth pipeline upward, we find a series of lines. To the left of a broken line, these curves represent turbulent flow in the transitional region between hdyraulically smooth and rough pipes in which both fluid viscosity and wall roughness are important. Wall roughness is a reactor which is determined by what type of pipe is used in the system. (Figure 11-4) gives some examples of the relative roughness.

The following equation can be obtained from experimental data for this portion of curves for commercial rough pipes.

$$\frac{1}{\sqrt{f}}-2.0\ \log_{10}\frac{r_0}{e}=1.74-2.0\ \log_{10}\left(l+18.7\frac{r_0/e}{\sqrt{fVd/\nu}}\right)$$

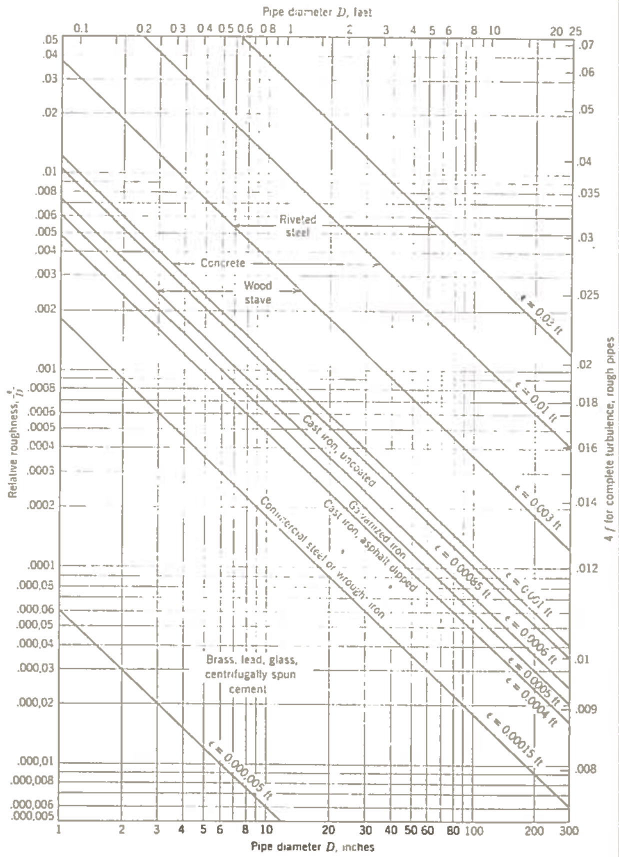
4. The critical zone is not easily defined by mathematical equations. Because of this, turbulent flow was assumed to exist after R = 2000. This means that values determined will be higher than those actually experienced with Reynolds numbers between 2000 & 5000.

ONE-DIMENSIONAL FLOW



Nikuradse's experimental results for pipes of uniform sand grain roughness.

(Figure 11-3)



(b) Relative roughness factors for new clean pipes. From *Pipe Friction Manual*, 3rd Edition. Copyright 1961 by the Hydraulic Institute, 122 East 42nd Street, New York.

(Figure 11-4)

ENTRANCE EFFECT ON PIPE

When flow goes from one component to another, a certain amount of swirl takes place. We consider this an entrance effect. The swirl decays in the downstream direction and the manner in which it has been investigated by B. Punnis. This swirl adds to the pressure drop already calculated in the tables. It almost completely dies out after an inlet length of 40 diameters at a Reynolds number of 1000. The additional pressure drop is:

ΔP Additional = K x ½ ρ Vo^2

Where ρ is the density, Vo is the initial velocity and K varies, but is less than unity.

PRESSURE DROP IN 90° BENDS FOR HOSE & TUBING

The pressure drop in bends can be assumed to be caused by three phenomenons:

1. the loss due to the curvature of the pipe.
2. the loss due to secondary flows.
3. the loss due to the length.

Factors 1 and 2 may be defined as a function of the radius of curvature and the velocity in the pipe, while factor 3 is just related to length. Much experimental work has been done to relate the resistance of 90° bends.

The factor to use for determining the pressure drop when using a standard bending tool (pressure drop will vary with different radius bends) is 4D = 12. This was computed for a r/d (radius of bend/diameter of tube) of 2.5, which is where the minimum pressure drops will take place.

LEAKS IN A SYSTEM

Most hydraulic leaks start at fittings — even when highly skilled craftsmen assemble systems — because fittings and adjacent line sections are most susceptible to vibration, motion and thermal shock. Vibration may be transmitted to lines and fittings from mechanical members, or it may originate from hydraulic shock within the lines. Whatever its cause, vibration eventually leads to cracked lines and loosened fittings. Excessive bending twisting or other distortion of the conductor wears and cracks lines, fittings and threads.

Thermal shock comes from sudden changes in fluid or ambient temperature, or exposure to thermal radiation, and causes rapid expansion or contraction of lines and fittings. Where dissimilar materials

are in contact, their differing expansion can open leak paths.

These leakage causes are usually embodied in one or more of the eight most common design errors:

1. Wrong conductor type. Pipe is often applied when a more flexible conductor, such as tubing or hose, should be used.
2. Wrong conductor material. Steel or stainless steel should be specified for high pressure lines, not aluminum.
3. Wrong conductor size. Undersized suction lines induce pump cavitation, producing excessive vibration. Undersized pressure lines may create excessive localized increases in heat.
4. Wrong wall thickness. Conductors with insufficient thickness are prone to burst or crack.
5. Wrong fitting type. Compression fittings, for example, are sometimes incorrectly specified for high-pressure lines.
6. Wrong material match. Fitting materials should match line materials.
7. Poor layout. Lines are often arranged so that fittings cannot be properly installed, or located so that accidental damage is likely.
8. Inadequate support or protection. Hydraulic lines must be supported at regular intervals and protected from heat and physical abuse.

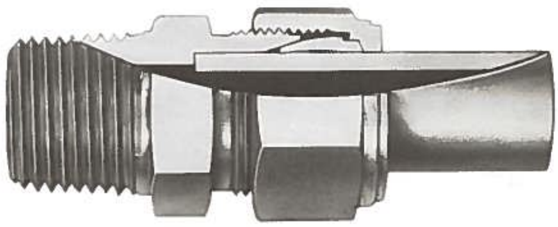
FITTINGS THAT DON'T LEAK

If a hydraulic system is to be free of leaks, fittings must be carefully chosen to match the type, size, wall thickness and material of the conductor, as well as the pressure, temperature, and environmental conditions of the application.

Fittings in the U.S. have either male or female tapered pipe threads or SAE straight threads. The most common materials are brass and steel.

Brass fittings are most commonly used with copper or plastic tubing for low or medium pressure applications, usually in relatively small sizes. Often, leaks result when cast-iron or steel pipe fittings with tapered threads are used with steel pipe on low or medium pressure applications. Steel fittings with steel tubing should be used for most high pressure applications, including most fluidpower applications.

Tube fittings are made of brass, aluminum, steel, and stainless steel. In general, the choice of material



(Figure 11-5)



(Figure 11-6)

is based on tube material: brass fittings with copper tubing, aluminum with aluminum, steel with steel, and stainless steel with stainless. Matching materials in this way provides a matched-strength assembly.

Tube fittings are either mechanical (unbounded) or bonded. Mechanical types include flared, bite, compression, and friction fittings. The two bonded types are welded and brazed fittings.

Welded or brazed fittings provide the maximum in sealing and holding power. They are ideal for hdyraulic lines handling very high pressure, and those that are hard to reach after installation. Selection of the proper type depends largely on the relationship between tube diameter and thickness, as shown in the selector chart.

Principal limitations on bonded fittings are the need for highly skilled welders and the comparatively long time required for assembly. However, some companies offer fittings that cut the need for assembly time and high labor skills because they are well suited to an automated welding process.

Among the mechanical tube fittings are compression, three-piece flared, friction, and bite fittings. Compression fittings have very limited hydraulic use, because their sealing and gripping powers are less than either flared or bite fittings.

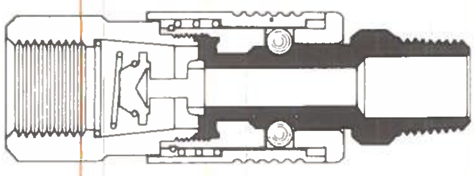
Three-piece flared fittings are generally considered to be the best general-purpose threaded design. They are suitable for a wide range of applications, including high pressures, high vacuum, very low or high temperatures, or severe vibration.

From the leakage point of view, their main advantages are generally better sealing and holding ability than either bite or compression fittings, and a lesser requirement for assembly and inspection skills than welded or brazed fittings. Three-piece flared fittings also cost less than other tube fittings when purchase and installation costs are considered.

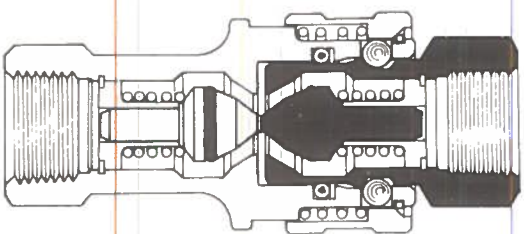
Bite fittings are generally used in high pressure applications for metal tubes that cannot accept flared fittings. Properly installed, bite fittings have good vibration resistance, holding strength, and sealing ability.

On the other hand, thin-wall tubing cannot accommodate the wedging forces generated by bite fittings. In addition, ferrules in bite fittings, need careful setting and checking before installation.

Hose fittings serve two functions: gripping the hose end, and attaching it to a port or another fitting. Carbon steel fittings are used for most applications,



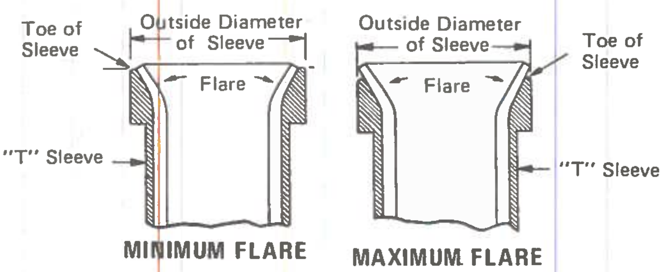
(Figure 11-7)



(Figure 11-8)



(Figure 11-9)



(Figure 11-10)

but stainless steel is used to cope with very high pressures or corrosive chemicals in the environment. Brass and monel fittings are available for special applications.

Permanent hose fittings are intended to be discarded with the hose, so they have not been as popular with many users as reusable fittings. Push-on fittings are generally limited to about 350 psi working pressure, so they are rarely used in hydraulic applications.

Reusable fittings can be readily removed from damaged hose and used again. For a given size, they have larger envelope dimensions than permanent fittings.

The most common reusable fittings are screwed on to the hose. They are suitable for working pressures ranging up to 5,000 psi and temepratures up to 200° F.

Clamped fittings are often specified for quick, easy field assembly, if space permits. Like screwed-on fittings, they are rated up to 5,000 psi and 200° F.

For plastic hose, the three-piece compression fitting is often the first choice. Working pressure ranges up to about 3,000 psi.

Segmented fittings are used mainly for large-diameter hose. They are suitable for working pressures up to 3,000 psi and temperatures to 250° F, but require special tools for field installation and removal.

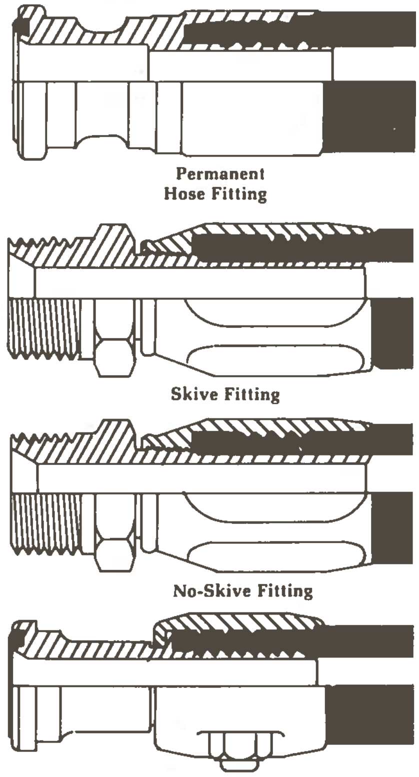
Quick couplings permit rapid connections and disconnections, and include shutoff valves to prevent loss of fluid when disconnected. Working pressures range to 5,000 psi, and seal materials include nitrile rubber and polyurethane.

Swivel adapters improve hose life by absorbing otherwise damaging twisting motion in the coupling. Working pressures range up to 3,000 psi and available seal materials include nitrile rubber, EPR, and flouroelastomer.

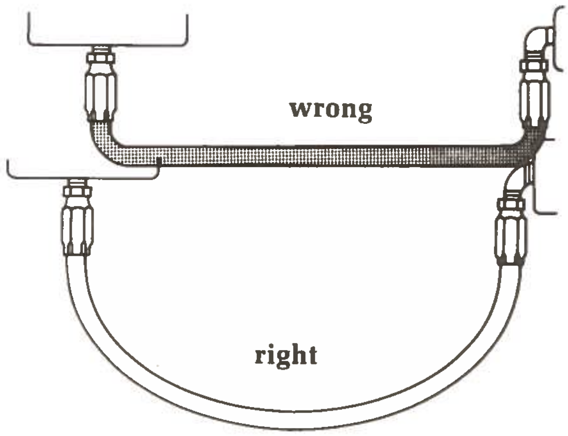
Because of their internal construction, both quick couplings and swivel adapters must be sized carefully to make sure that adequate flow capacity is maintained.

WHERE TO PUT LINES

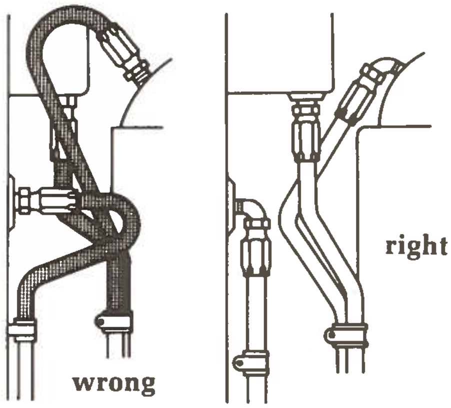
Hydraulic lines that are properly selected and sized can still leak if they are not skillfully laid out, supported, and shielded. Even the best of lines are vulnerable to damage from poor routing or protection.



(Figure 11-11)



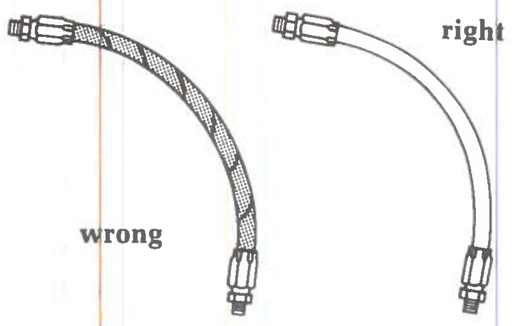
Ample bend radius should be provided to avoid collapsing of line and restriction of flow. Exceeding minimum bend radius will greatly reduce hose assembly life. **(Figure 11-12)**



Use elbows or other adapters as necessary to eliminate excess hose length and to insure neater installation for easier maintenance.

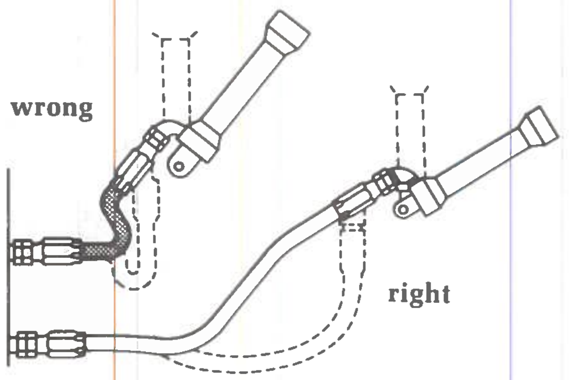
(Figure 11-13)

right vs. wrong



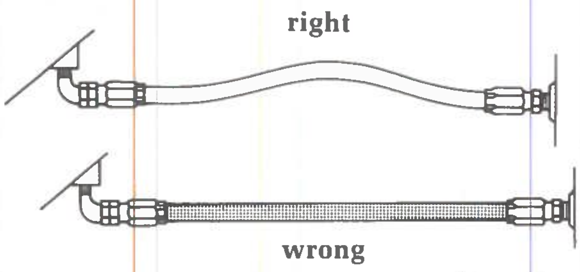
Hose is weakened when installed in twisted position. Also, pressure in twisted hose tends to loosen fitting connections.

(Figure 11-14)



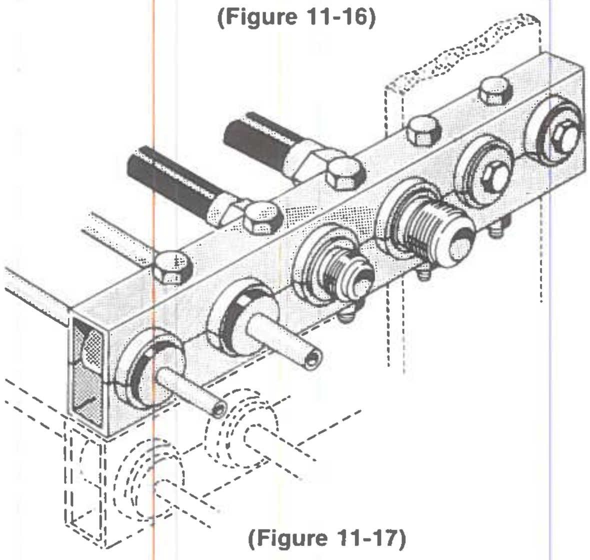
When hose assembly is installed in a flexing application, remember that metal hose fittings are not part of flexible portion. Allow ample free length for flexing.

(Figure 11-15)



Pressure can change hose length as much as +2% or -6%. Provide slack in line to compensate for hose length changes.

(Figure 11-16)



(Figure 11-17)

Tubing has become practically the universal hydraulic conductor in applications where components have no relative motion. But to take best advantage of tubing, runs should have the smallest possible number of joints. Where angles are required, use bend rather than fittings.

Connectors on long tubing runs are particularly prone to leakage because of strains induced by tube weight and whipping from hydraulic shock. These strains can be largely eliminated by proper support.

The most critical support point is about 5 to 6 in. behind the fitting. If vibrations are damped here, their effect is negligible. Support is also needed at points where valves and gages are inserted in the line, and where tube runs join hose runs. Additional required support points are shown in the table.

Almost as important as support is tube layout. Many lines leak simply because they are positioned where they interfere with the machine operator's movements, block access doors or high-maintenance parts or project into areas where they can be damaged by accident. Where several lines run together in the same direction, group and protect them with multiple clamps or tube runs.

Hose is less susceptible to damage from vibration, deflection, torque or impact than are pipe or tube, but it is more vulnerable to damage from abrasion, pull-out, and heat. Because of this vulnerability, hose guards should be used near moving machine elements, and insulation should be added near hot surfaces. Brackets and clamps are needed to keep hose and insulators in place.

In runs between moving members, allow enough hose length so that fittings are not subjected to high pullout forces when the members reach their limits of travel. Even if hoses are not pulled out of fittings, stretching the hose restricts flow.

For long straight runs, provide about 5% extra hose length to allow for hose contraction during pressure surges, and provide enough hose length for wide-radius curves. Most manufacturers give catalog listings of minimum bend radius, if bends must be sharper than the minimum radius, use angle fittings instead of bends.

KEEPING PORTS TIGHT

Hydraulic components — including manifolds, pumps, motors, cylinders and valves — have inlet, outlet, and control ports that are either threaded or flanged to receive fittings or conductors. Choice of port design affects not only first cost and ease of assembly, but also the likelihood of leakage.

SAE straight thread fittings with o-ring seals are recommended for the majority of component port connections to achieve leak-free joints.

Straight-threaded fittings require lower tightening torques for proper installation than do tapered threads. This feature eliminates the danger of distorting or cracking component housings, which often happens with tapered-thread fittings. Straight-thread fittings are not so susceptible to vibration as tapered thread fittings, which shake loose and leak. Furthermore, the threads on a straight-thread fitting do not deform during installation and the fitting can be reused many times. Reuse of tapered-thread fittings is limited, and they require a sealant that often gets into the system when fittings are improperly installed.

Flanged ports are often the best choice when ports and flow are large, or pressure and temperature conditions are critical. These ports are sealed with an o-ring compressed against a ground surface immediately around the port. When the port mates with pipe or tubing, a stem carrying the o-ring is welded in place. With hose, the stem may be clamped, rather than welded.

CONCLUSION

When selecting a conductor for a particular system, it is a good rule to keep pressure lines flow velocities below 15 ft/sec., return line flows below 8 ft/sec and suction line flows below 4 ft/sec. Sharp bends and fittings should be kept to a minimum.

Designing against hydraulic leaks is generally a fairly straightforward task, but it depends on coordinating design decisions for a great many discrete components that may not be obviously related. If your system design meets these criteria, you can logically expect it to be leak-free.

Ports:

Do they all have same thread or flange type?

Are they spaced and located so assemblers can tighten fittings?

Are o-rings or sealants compatible with fluid?

Fittings:

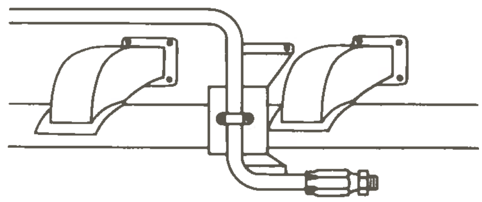
Are materials compatible with component bodies, conductors, fluid and environment?

Are assemblers familiar with this type of fitting?

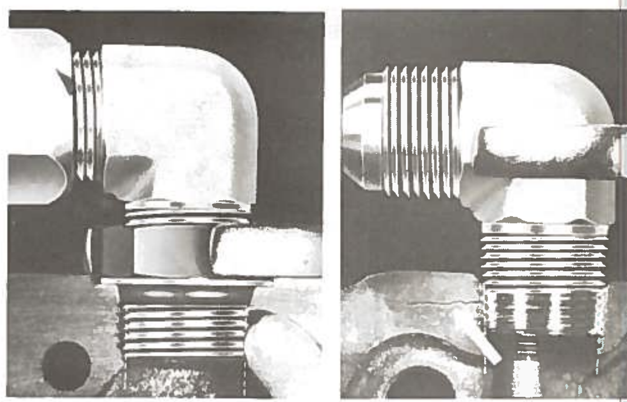
Is pullout strength adequate?

Are pressures and temperatures within rated limits?

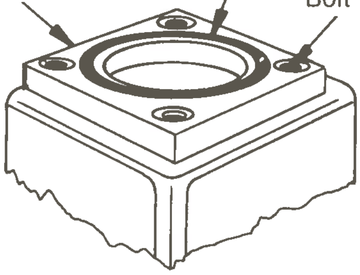




(Figure 11-18)



(Figure 11-19)



(Figure 11-20)

Will remakes damage fittings or other components?

Plumbing layout:

Are joints kept to a minimum?

Are valves and gages separately mounted?

Are runs properly supported by clamps and hangers?

Are runs protected from accidental damage?

Do bends have proper radius?

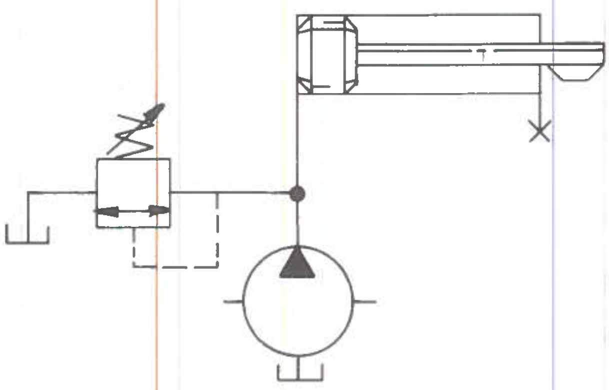
Does layout allow easy assembly?

ANALYZING HYDRAULIC SYSTEMS EXERCISES

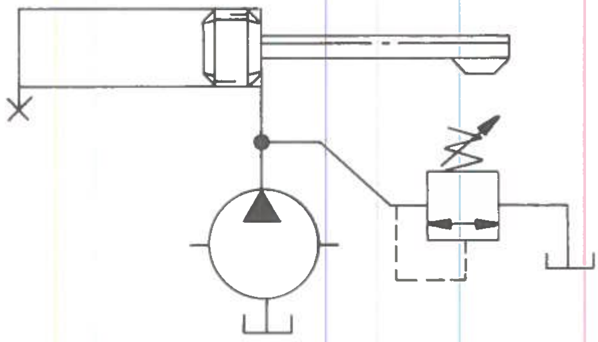
ASSIGNMENT #1

1. A cylinder is used to clamp a part with a force of 10,000 lbs. The maximum working pressure is 1000 psi. What is the minimum size standard cylinder that would do the job?? What is the cylinder part number??

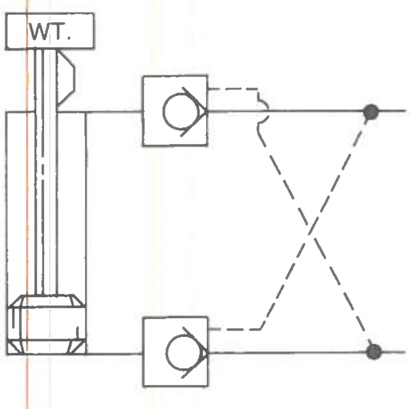
2. In order to check the leakage across the piston of a cylinder, Technician #1 connects the cylinder as shown below. The rod extends and the speed is monitored.



Technician #2 says that he could have hooked the cylinder as shown below and obtained the same results by monitoring the speed of the retracting rod. Is the Technician correct??



3. A hydraulic lift is used to raise concrete from the first to the second floor of the building. The load is 50,000 lbs. and the cylinder has a diameter of 8 inches. The rod diameter is 3.25 inches. Because of safety considerations, load checks (with no leakage) keep the load from accidently falling. Over a weekend the cylinder is left at mid-stroke with a load of concrete. Upon returning to work on Monday, the foreman finds the rod seal blown out. Why did this happen??



4. A cylinder with a 5-inch bore and a 36-inch stroke, extends in 3 seconds. It moves a 25-ton load.

   a. How much HP is developed??

   b. What is the pressure needed to extend the cylinder??

   c. What GPM must be delivered to the cylinder to move it at its required speed??

5. Consider the following system::

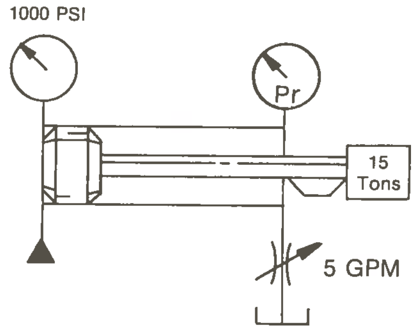
   7″ bore
   4″ rod

   a. What value is found at Pr??

   b. What flow enters the cylinder??

   c. What is the rod speed??

   d. What horsepower is developed??



ASSIGNMENT #2

1. An application exists for the cylinder pictured below. The 60-ton weight is to be accelerated to 480 inches per minute in .1 seconds. It must overcome the static coefficient of friction of .3. The stroke of the cylinder is 40 inches. The maximum relief valve setting that can be used is 2500 psi.

   The load is also to be decelerated with a standard cushion ($1\frac{5}{16}$ inches in length). However, when deceleration takes place, there is no friction.

   The system is to be designed for minimum metal: (smallest cylinder available)

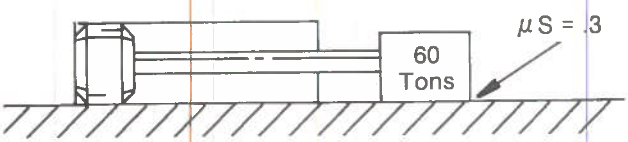
   a. What size cylinder did you select?? Why?? Any accessories??

   b. What is the maximum pressure developed in the cylinder??

   c. What size pump is needed?? (GPM)

   d. What is the relief valve setting??

   e. State the complete cylinder part number.



ASSIGNMENT #3

1. You are requested to design a 50-ton press using one cylinder, the shown directional control valve and pump and limit relief pressure to 2500 psi maximum.

   The cylinder is to extend with no load (about 100 psi) for the first 5 inches of the travel. During this period of time, the directional valve is centered and the cylinder is in regeneration. Its rod speed is to be 2 in/sec. For return, the rod speed is approximately 1 in/sec with 300 psi at rod end.

   Pressing speed is determined by the pump selected for regeneration and return.

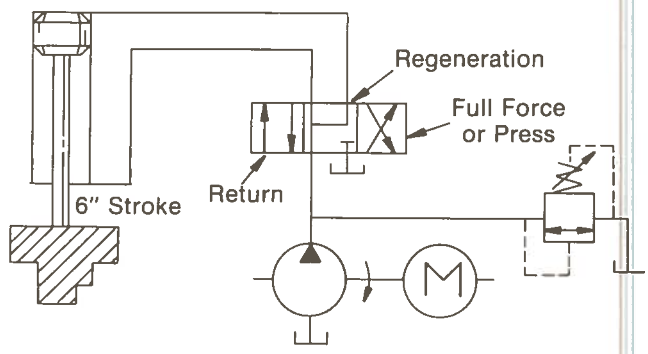
   a. What cylinder is selected?? What is the cylinder part number??

   b. What pump is selected??

   c. What is the relief valve setting??

   d. What is the rod speed — (1) in pressing??
   (2) in return??

   e. What is the HPrms?? (consider the pump 100% efficient)



2. A hydraulic motor is used to lift a 200 lb. weight by coiling wire around an 8-inch diameter drum. The speed of the motor is 800 RPM's.

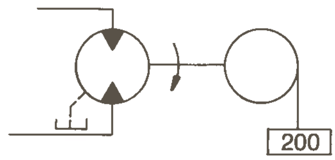
   a. What is the relief valve setting??

   b. What is the pump flow rate??

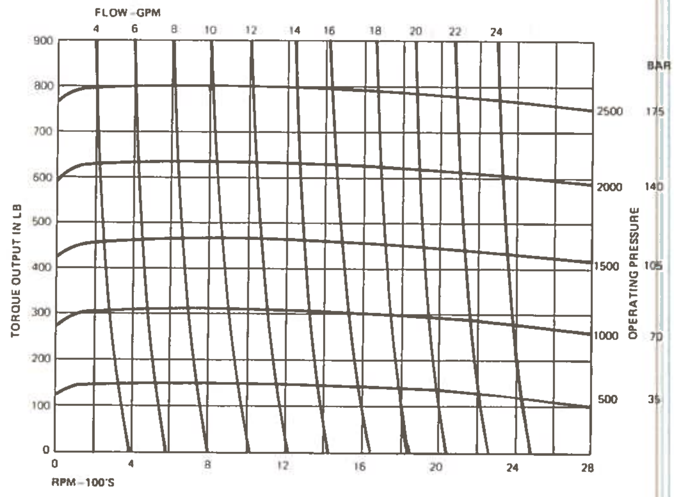
   c. What is the efficiency while running??

   d. How much heat is generated (in HP)??

   e. What is the motor part number??



810 SERIES, F CARTRIDGE
34.9 IN.-LBS/100 PSI, 2.20 CU. IN./REV.



ASSIGNMENT #4

1. A maintenance foreman states he has the following motor:

   "We have a 6 pole, Nema B. TEFC motor."

   What does he mean by each one of the three previous terms??

2. The cycle of a particular machine is as follows:

   a. 25 hp for 10 sec.

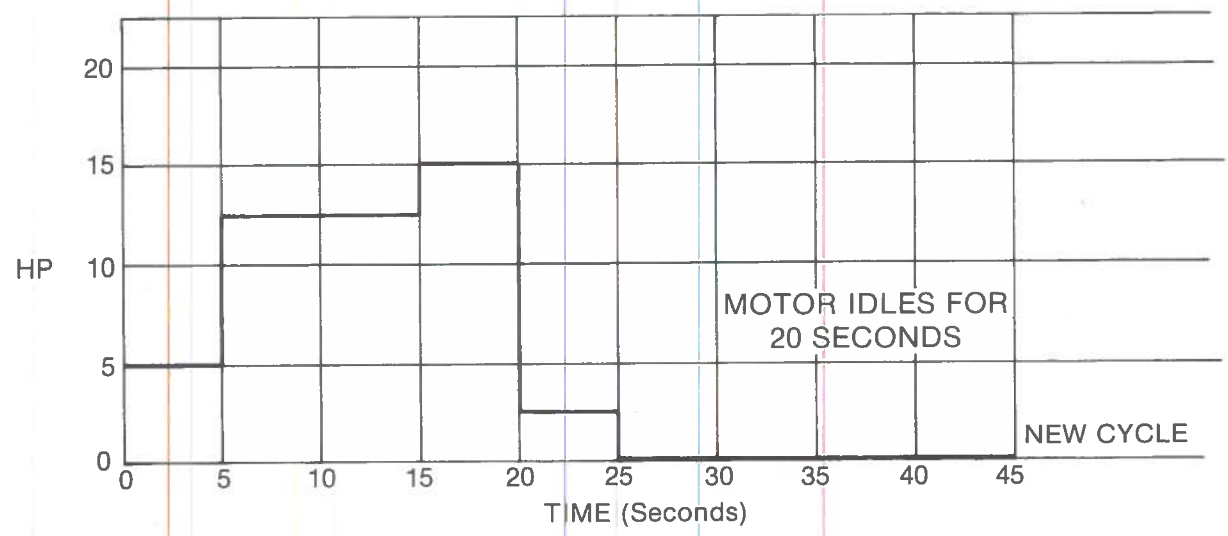
   b. 17 hp for 6 sec.

   c. 30 hp for 8 sec.

   d. 2 hp for 15 sec.

   What size electric motor should be used for this application??

3. For an open motor, the horsepower cycle is shown below:



   What is the HPrms for the cycle??

ASSIGNMENT #5

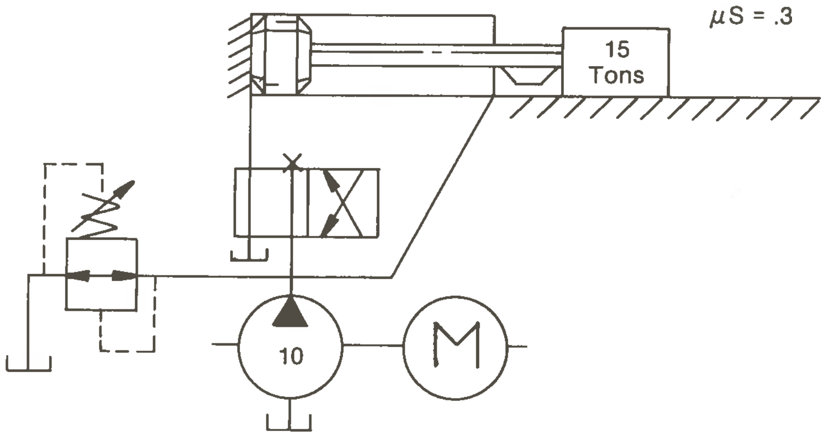
1. A 5″ bore, 2″ rod cylinder is used in regeneration. The circuit for the cylinder is shown below:

   a. What pressure is needed to extend the load??

   b. What is the rod speed in extension??

   c. What is the input horsepower??

   d. How much flow must the directional valve pass in regeneration??



2. The restriction is of such value that the pump can only put 8.1 of its 15 GPM into the cylinder. A gage at the cap end reads 1500 psi.

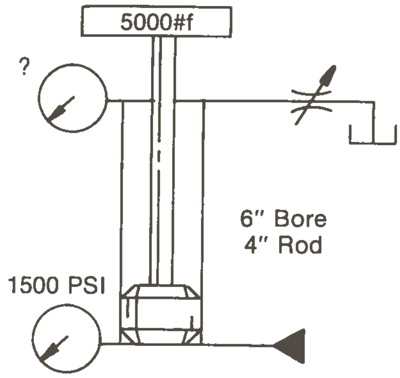
   a. What is the restriction set for (in GPM)??

   b. What would a gage at the rod end read??

   c. What is the rod speed??

   d. What is the horsepower out of the cylinder??

   e. What is the heat generated in the above circuit??



3. The following chart shows the horsepower cycle of a system. What size motor would you suggest??

HP	Time (Sec.)
12	7
22.5	10
5	15
2	5

ASSIGNMENT #6

1. Eric Carmen decided to use a Parker PVV142 Series 65 GPM Variable Volume Vane Pump at his plant situated in Denver, Colorado. He is to be operating at 1200 RPMs. What is the maximum he may read at the gage??

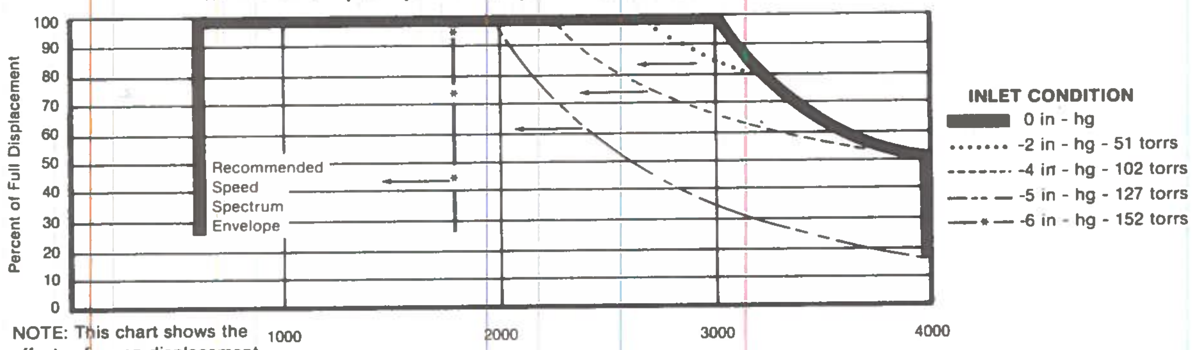
   INLET CONDITIONS:
   Not to exceed 7 in. Hg at 1200 RPM or 5 in. Hg at 1800 RPM.

2. At 1000 ft. above sea level, a particular power unit is reading 6.1 inches of Mercury (in.Hg). The pump's maximum allowable vacuum is 5 in. Hg. The owner says he needs to read 1.1 in.Hg. extra because he has 1.1 in.Hg. less than a site at sea level. Is he correct in his previous statement?? If not, why??

3. Two Vardis VP033's are used at 2000 ft. above sea level at 100% displacement and 1725 RPMs and 2000 RPMs. What will the maximum vacuum gage readings be at the respective reservoir suction lines??

(Test Conditions — Oil 100 SSU 120° F (49° C))

Recommended Speed Spectrum for Specific Inlet Conditions (VP033)



NOTE: This chart shows the effects of pump displacement and inlet conditions on recommended speed limits

ASSIGNMENT #7

1. An "E" Series 410 pump is operated at the following pressures and times. (1800 RPM's)

PSI	Time (Sec.)
400	8
100	14
1500	7
2000	5

What is the RMS horsepower?? What size electric motor do you need??

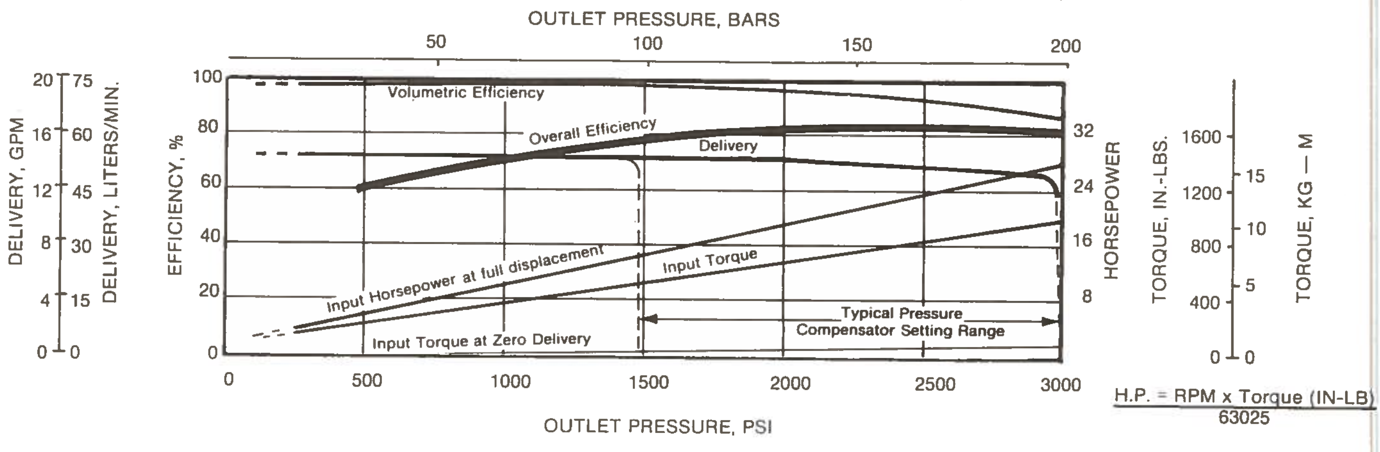
Cartridge Letter	Nominal Delivery		Pressure		Delivery				Input HP	
					1200 RPM		1800 RPM		1200 RPM	1800 RPM
	GPM	L/Min.	PSI	BAR	GPM	L/Min.	GPM	L/Min.		
E	5	18.9	100	7	5.3	20.0	8.0	30.2	0.4	0.6
			1000	70	4.5	17.0	7.3	27.6	3.9	5.8
			2500	172	3.6	13.6	6.4	24.2	8.5	12.7

2. A Vardis pump VP033 is used in the following cycle. Its compensator setting is 2500 psi. What is the RMS horsepower?? What size electric motor do you need?? (1800 RPM's)

Pressure	Time (Sec.)	Flow (GPM)
1500	10	?
2000	5	?
1750	7	?
?	8	10
2500	30	0

Performance Characteristics

(Test Conditions: Oil 100 SSU 120° F (49° C); 1725 rpm; full displacement; atmospheric inlet)



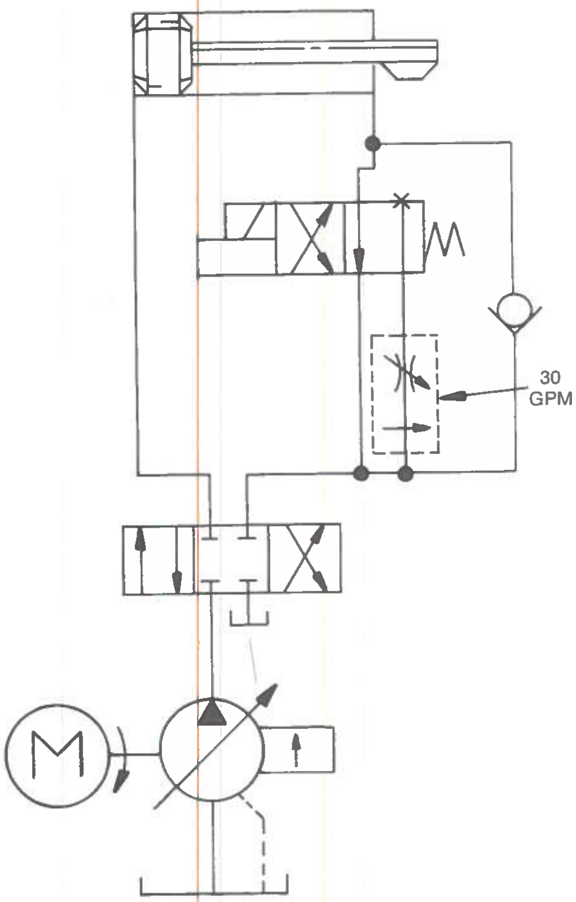
ASSIGNMENT #8

1. A cylinder with an 8″ bore, 3½″ rod and 15″ stroke is used in the following circuit. It is a rapid advance and feed type circuit whereby when a limit is made at 10″ extension of its stroke, the 2 position 4-way valve is energized and a flow control (set for 30 GPM) enters the circuit. During retraction, full flow can pass through the check. The idle time is 5 seconds.

   The pump employed in this circuit is a PVV142 operated at 1200 RPMs. The compensator is set at 2000 PSI.

   Neglecting all inefficiencies of the components, except for the pump, what size electric motor would you prescribe for this circuit.

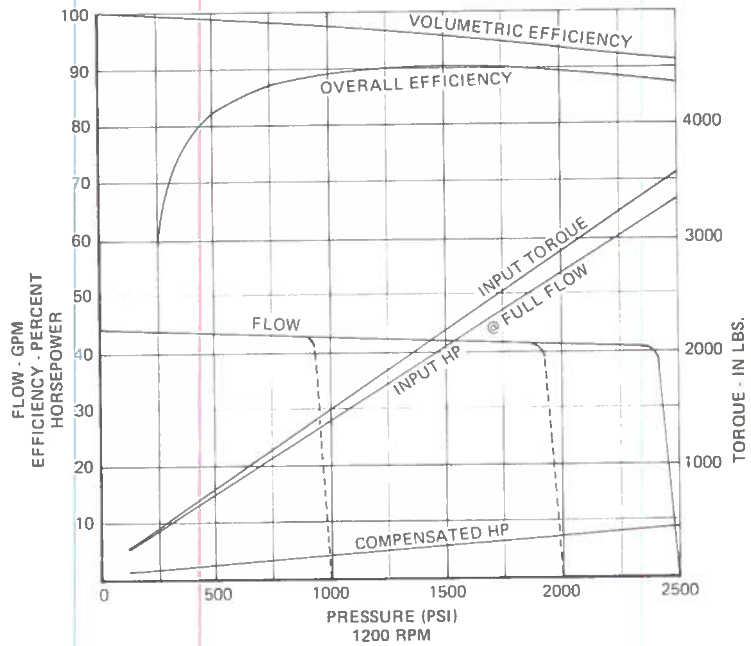
   A force - time chart of the circuit is as follows:



Cycle	Force	Flow	Stroke
Rapid Extension	75,400	Full	First 10 inches
Feed	90,000	Set by flow con.	Last 5 inches
Return	40,630	Full	15 inches
Idle		0	No Movement

   a. At various times in the cycle, what are the pump's . . .
      1.) pressures?? (Pc & Pr)??
      2.) flow rates??
      3.) horsepower (input from electric motor)??

   b. What is the HPrms?? What is the electric motor size??



2. A series of gear pumps have the following suction specification:

   Maximum allowable vacuum is 7 in. Hg. when operated with a fire resistant fluid.

   A manufacturer of power units using the above pump in Marco Island, Florida, (at sea level), wants to know the maximum elevation that he may ship his power units if he reads 1.8 in. hg. at his test site?? (This value of 1.8 in. Hg. is read at a vacuum gage situated at the pump's inlet port.)

3. An 810 motor with an "F" cartridge is needed to be run at 1500 RPM's with a running torque of 400 in.pounds. The starting torque for the system is 800 inch pounds of torque.

   What is the:

   a. Flowrate into the motor??

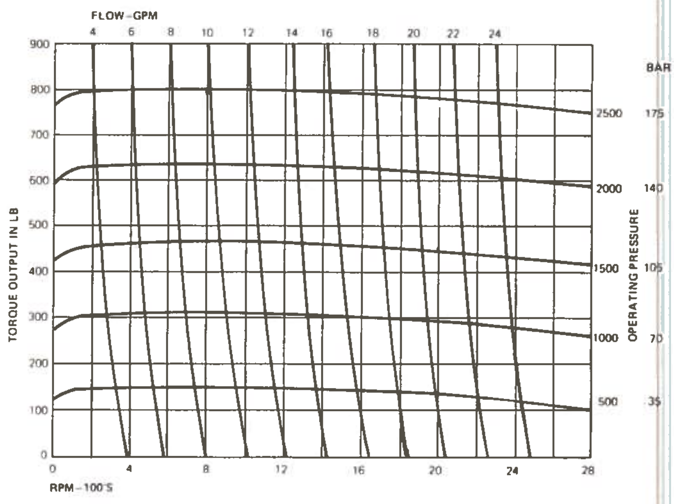
   b. Running pressure??

   c. Relief valve setting??

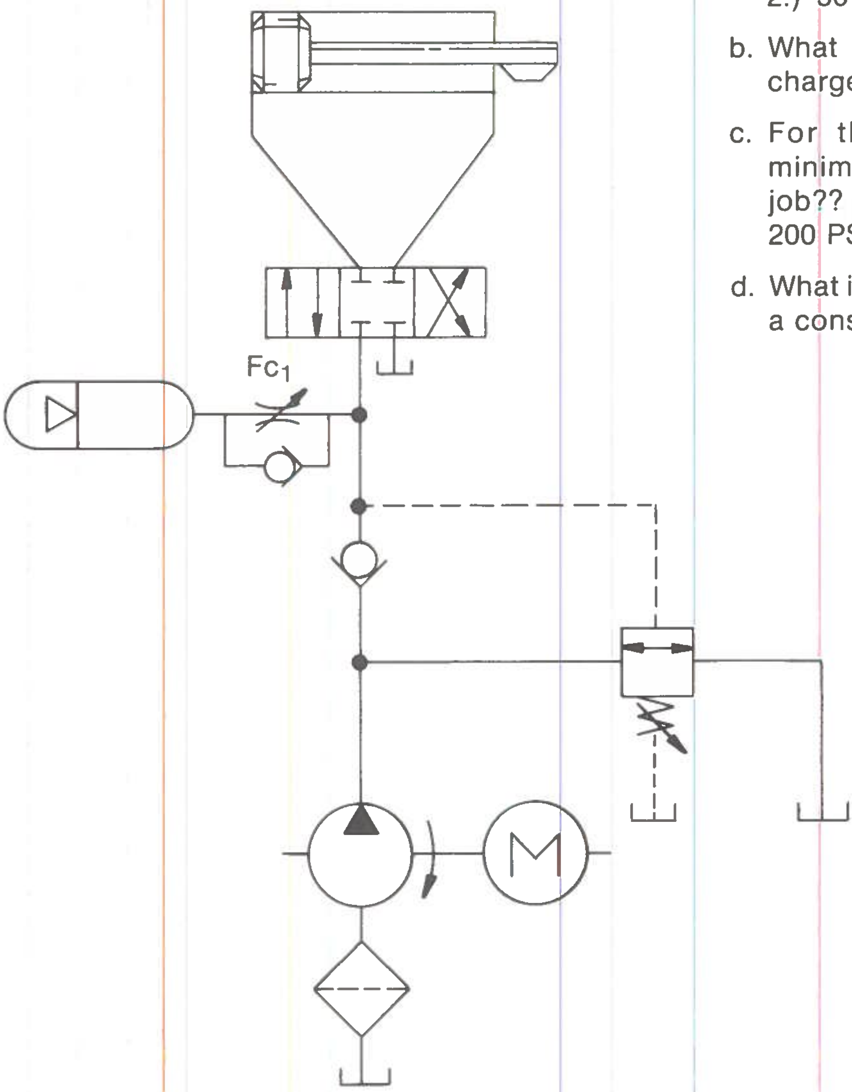
   d. The heat generated during running??

4. Consider a cylinder with a piston area of 10 sq. inches and a rod area of 5 sq. inches. It is powered by a 10 GPM pump with the relief valve set for 1200 psi. The load is 20,000 lbs. with a coefficient of friction of .5. How long will it take to accelerate the load to the pump's full flow velocity?? (Answer in seconds)

810 SERIES, F CARTRIDGE
34.9 IN.-LBS/100 PSI, 2.20 CU. IN./REV.



ASSIGNMENT #9



1. A 4-inch bore with a 50-inch stroke has to be moved at a rate of 120 in/min. Because of the mechanical advantage of the linkage the cylinder must only produce ⅓ the force necessary to move 50 tons over a surface with a coefficient of friction of .15. An accumulator will assist the 5 GPM pump to achieve the necessary rod speed. (1¾″ rod diameter)

   a. Specify the necessary volume size, precharge and maximum pressure for an accumulator designed for:

      1.) minimum metal (What is the efficiency??)

      2.) 50% efficient operation

   b. What is the minimum idle time needed to charge the accumulator??

   c. For the previous idle time, what is the minimum HP motor (standard) that will do the job?? (use RMS criteria) Return pressure is 200 PSI, consider pump 100% efficient.

   d. What is the flow control (FC_1) set for to achieve a constant rod speed??

ASSIGNMENT #10

1. A cylinder with a 4" bore and 20-inch stroke is needed to be run at a rod speed of 250 in/min. Because of the requirements of the system, a pump of 5 GPM plus an accumulator is decided upon. The accumulator is designed for 60% efficiency. The load for the cylinder is 2512 lbs.
   a. What size standard accumulator was selected for this circuit?? What is its precharge??
   b. What horsepower does it take to charge the accumulator??
   c. What is the maximum pressure that the accumulator is charged to by the pump??
   d. What is the minimum idle time needed to charge the accumulator??
2. A D31W valve is used in the following circuit:

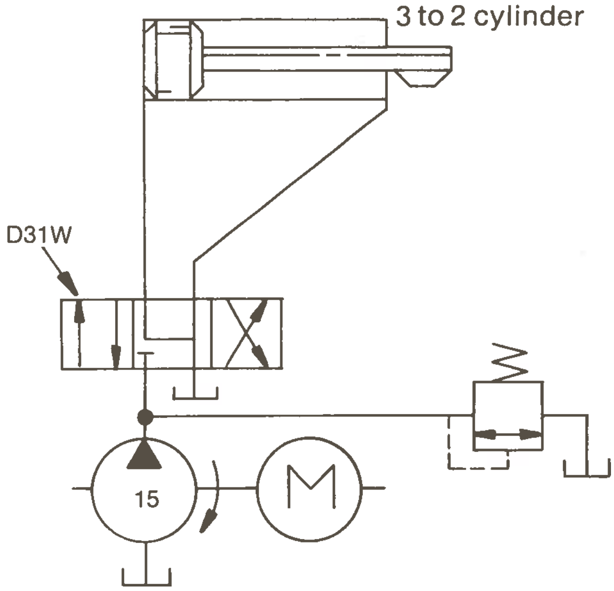


Chart "A" Curve Reference					
Max. Flow	Curve Number				
	P-A	P-B	P-T	A-T	B-T
30	5	5		6	6

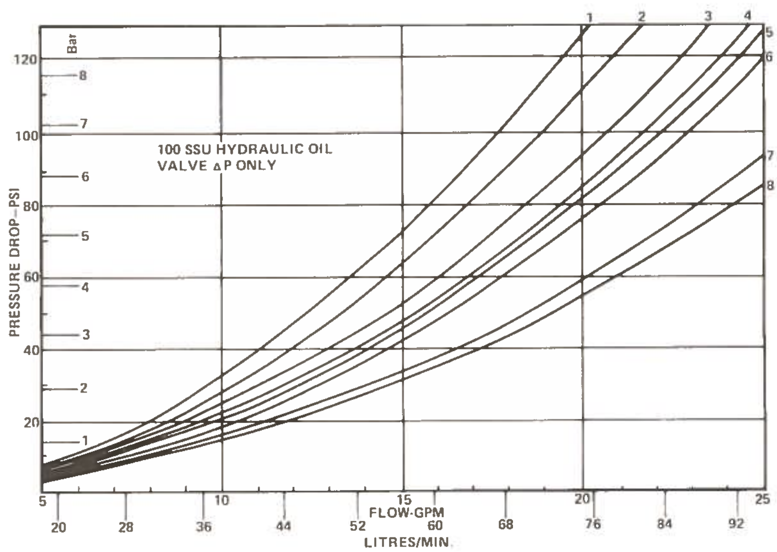
Fill in the following chart:

	P-A	B-T	P-B	A-T	P-T
Pressure Drop					

What is the heat generated in:

a. extension??

b. retraction??



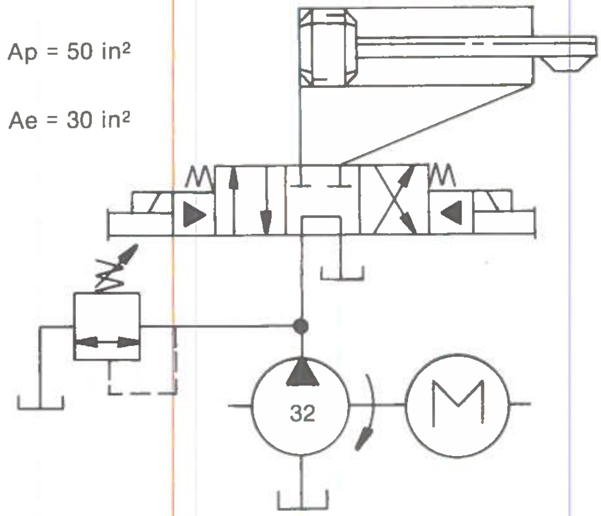
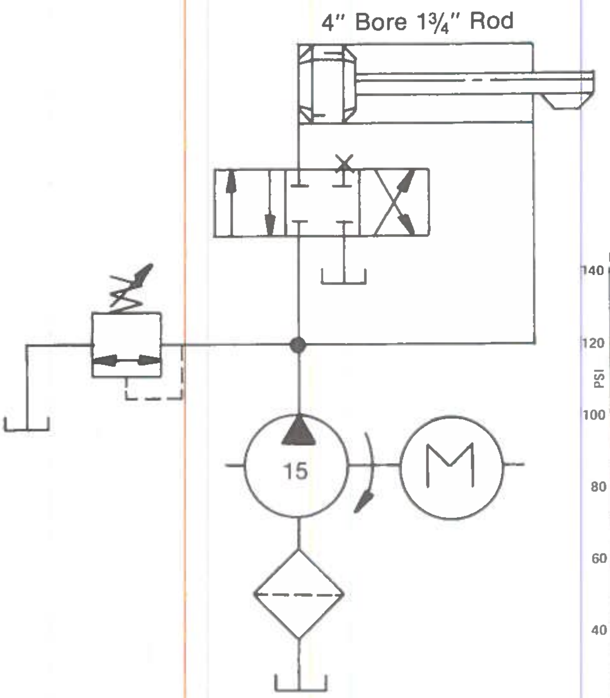


CHART "A" CURVE REFERENCE						
SPOOL CODE	MAX. FLOW *	CURVE NUMBER				
		P-A	P-B	P-T	A-T	B-T
04	60	3	3	6	3	5



D103W

SPOOL CODE	MAX. FLOW	CURVE NUMBER				
		P-A	P-B	P-T	A-T	B-T
01	110	4	4		2	3

ASSIGNMENT #11

1. A Parker D63W directional control valve is used in the following circuits:

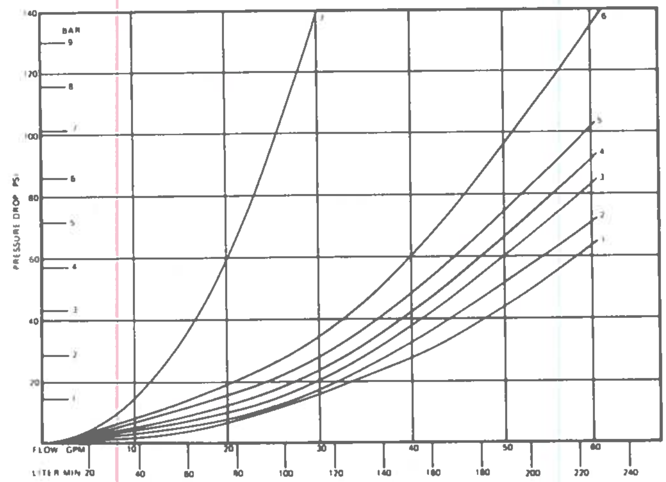
   What are the pressure drops between the various ports??

   What is the heat generated during:

   a. idle??

   b. forward??

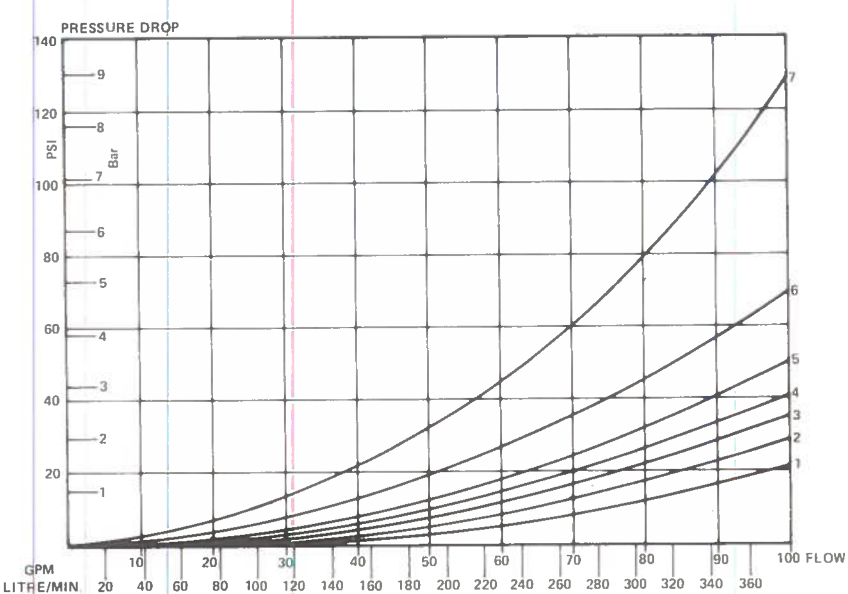
   c. return??



2. What size directional control valve would you select for the following circuit?? Give the part number. What is the heat generated during:

   a. forward??

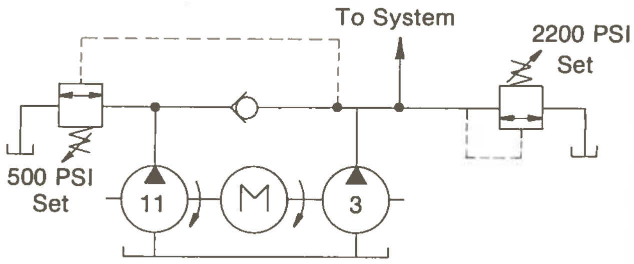
   b. return??



ASSIGNMENT #12

1. We have the following Hi-Lo system. It consists of a 412HB21R1AB pump. Fill in the following chart: (1200 RPM) (11 GPM/3 GPM)

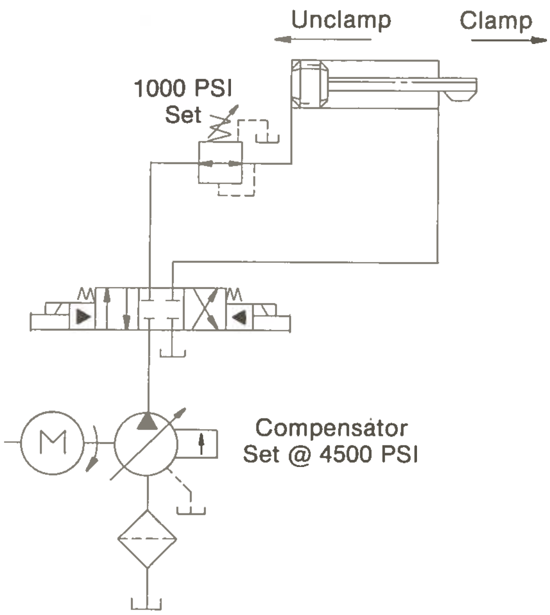
Cycle	System Pressure	GPM To System	Input HP
1	300		
2	1600		
3	2000		



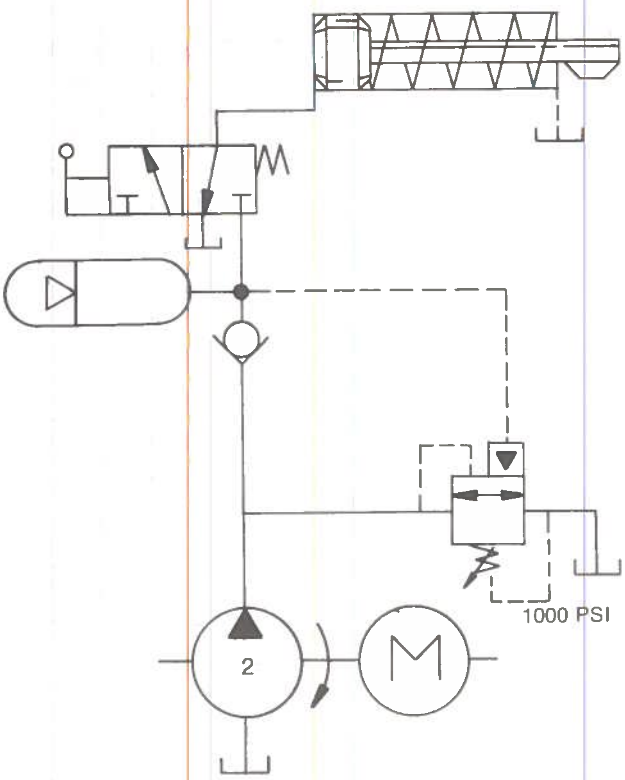
Cartridge Letter	Nominal Delivery		Pressure		Delivery 1200 RPM		Input HP
	GPM	L/Min.	PSI	BAR	GPM	L/Min.	1200 RPM
B	3	11.3	100	7	3.3	12.5	0.3
			1000	70	2.5	9.5	2.6
			2500	172	1.6	6.0	5.6
E	5	18.9	100	7	5.3	20.0	0.4
			1000	70	4.5	17.0	3.9
			2500	172	3.6	13.6	8.5
F	8	30.2	100	7	8.4	31.8	0.5
			1000	70	7.6	28.7	5.9
			2500	172	6.7	25.3	13.3
H	11	41.6	100	7	11.4	43.1	0.8
			1000	70	10.6	40.1	7.8
			2500	172	9.7	36.7	17.8

Maximum RPM for all sizes is 2000 RPM.

2. A pressure reducing valve is used in the following system. It is a 1¼ valve with a drain flow of ½ GPM. What is the heat generated when the clamp is holding??



ASSIGNMENT #13



1. The following system is used for clamping. The cylinder has a leakage rate of 10 in^3/min. past the seals; the differential unloading relief valve is #MURN 16M 3AA set for 1000 PSI. The accumulator, A2A0058A1 with a precharge of 500 PSI is used to hold pressure. How long does the pump stay:

   a. loaded??

   b. unloaded??

Code No.	Range P.S.I.	Pilot Head No.	Cut-In (% Of Setting)
1	800 to 3000	MA 678411	88%
3	400 to 1000	MA 678557	70%

MODEL A2A0058A1

GAS PRE-CHARGE PRESSURE-PSI (GAGE)	OPERATING PRESSURE — P S I (gage)														
	100	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
200			14.2	22.1	27.5	31.2	34.0	36.3	38.3	39.8	41.0	42.1	43.0	43.8	44.6
			18.9	*28.6*	*34.6*	*38.7*	*41.6*	*43.8*	*45.5*	*46.8*	*48.0*	*49.0*	*49.8*	*50.5*	*51.1*
300				10.7	17.1	22.5	25.9	29.2	31.5	33.5	35.2	36.6	38.0	39.0	40.0
				14.5	*23.2*	*29.1*	*32.9*	*36.5*	*39.0*	*41.1*	*42.8*	*43.5*	*45.4*	*46.3*	*47.1*
400					8.45	14.5	19.2	22.7	25.5	28.0	30.0	31.8	33.2	34.6	35.8
					11.5	*19.4*	*25.0*	*29.2*	*32.7*	*35.2*	*37.4*	*39.3*	*40.8*	*42.2*	*43.3*
500						7.12	12.4	16.6	19.9	22.7	25.1	27.2	28.9	30.4	31.8
						9.70	*16.7*	*21.9*	*26.0*	*29.3*	*32.0*	*34.3*	*36.2*	*37.9*	*39.3*

GAS PRE-CHARGE PRESSURE-PSI (GAGE)	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900	3000
200	45.3	46.0	46.4	46.9	47.3	47.8	48.1	48.5	48.9	49.0	49.4	49.7	49.8	50.1	50.4
	51.6	*52.1*	*52.5*	*52.8*	*53.1*	*53.5*	*53.7*	*54.0*	*54.3*	*54.4*	*54.6*	*54.8*	*54.9*	*55.1*	*55.2*
300	40.8	41.6	42.4	43.0	43.6	44.1	44.6	45.0	45.5	46.0	46.3	46.5	46.8	47.2	47.6
	47.9	*48.6*	*49.2*	*49.8*	*50.2*	*50.7*	*51.1*	*51.4*	*51.8*	*52.1*	*52.4*	*52.6*	*52.9*	*53.1*	*53.4*
400	36.8	37.9	38.4	39.5	40.1	40.8	41.4	41.9	42.4	43.0	43.4	43.8	44.3	44.7	44.9
	44.3	*45.2*	*45.9*	*46.7*	*47.2*	*47.8*	*48.4*	*49.0*	*49.4*	*49.7*	*50.2*	*50.4*	*50.7*	*51.0*	*51.4*
500	33.0	34.2	35.0	36.0	36.8	37.7	38.4	39.2	39.6	40.3	40.8	41.5	41.7	42.0	42.5
	40.6	*41.7*	*42.7*	*43.5*	*44.3*	*45.0*	*45.7*	*46.4*	*46.9*	*47.4*	*47.8*	*48.2*	*48.7*	*49.0*	*49.4*

ASSIGNMENT #14

1. For both systems below, answer the following questions.

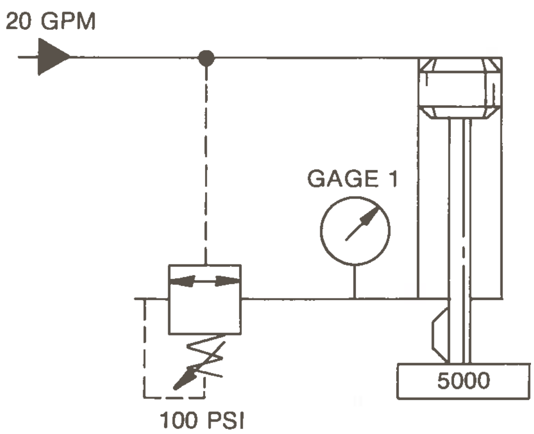
   a. With the press stationary, what does gage 1 and 2 read??

   b. With the press moving, what does gage 1 and 2 read??

   c. What is the maximum pressing force with the relief set for 1000 psi??

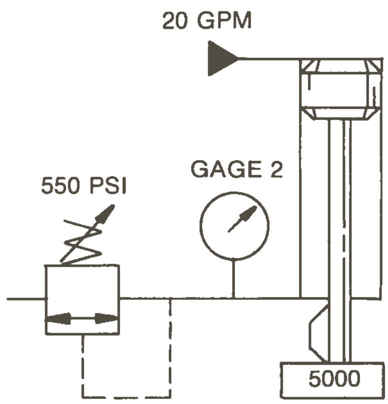
   d. What is the heat generated during extension??

SYSTEM #1



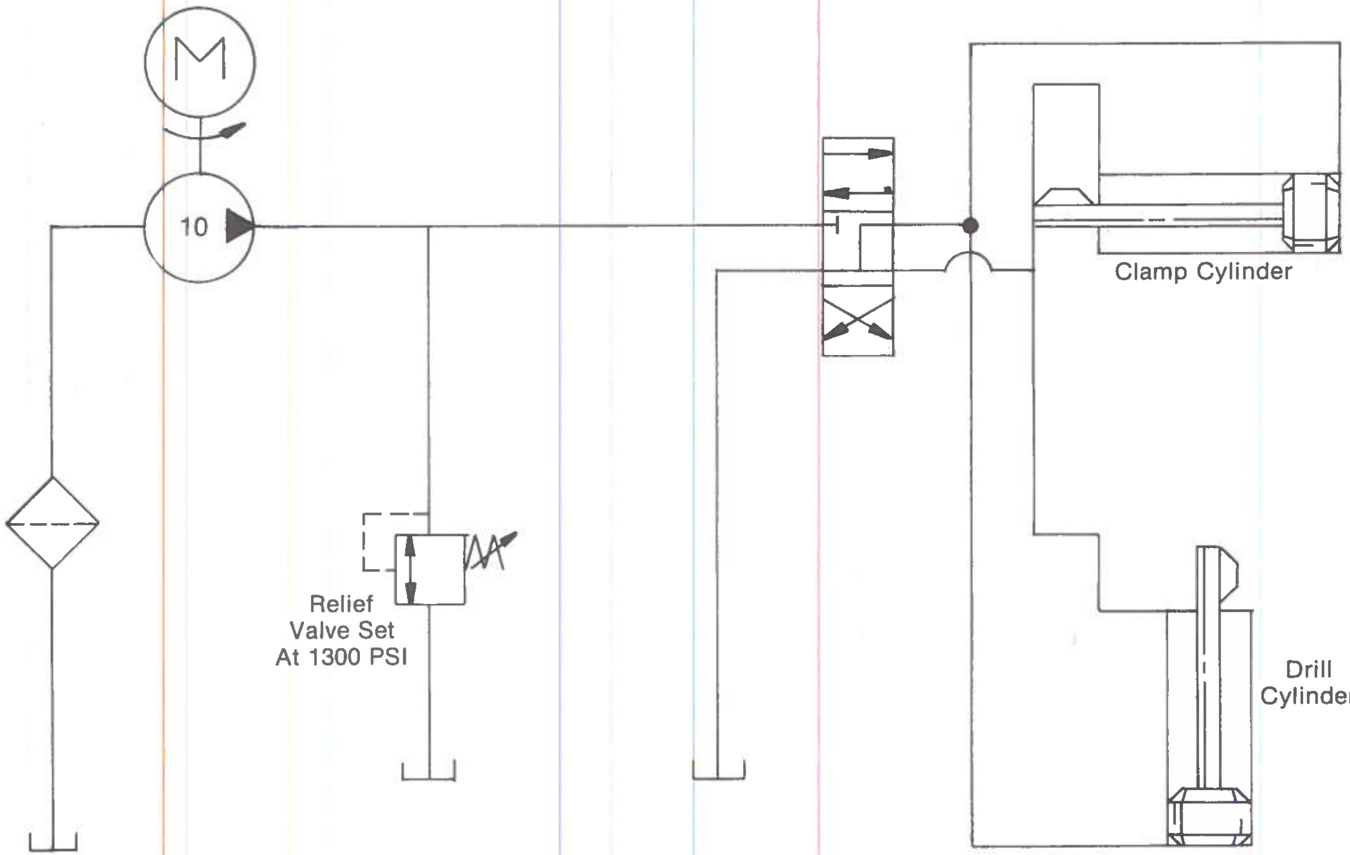
$A_p = 20$ in^2
$A_e = 10$ in^2

SYSTEM #2



ASSIGNMENT #15

1. The following circuit exists consisting of a clamp and drill. The clamp must extend and clamp at 1000 psi before the drill may drill. Insert the correct valve to accomplish such an action. When the drill is extending, what is the heat generated if the drill cuts metal at 300 psi??



ASSIGNMENT #16

1. A D63W series valve with the following part # (D63W1D4Y13) is being used. Because of system conditioning, the valve must have external pilot and drain. Explain what it takes to convert the valve. Is there any mounting restriction on the valve?? What should the valve mounting bolts be torqued to and in what sequence??

D | 63 | W | ☐ | ☐ | ☐ | ☐ | ☐ | ☐ | 13

DIRECTIONAL CONTROL VALVE — D

BASIC VALVE SIZE

63	3/4" WITH 3/8" PILOT OPERATOR

ACTUATOR

W	WET ARMATURE SOLENOID

PILOT SUPPLY & DRAIN

CODE	SUPPLY	DRAIN
1	INT	EXT
2	EXT	EXT
3	BACK PRESS CHECK	EXT
4•	INT	INT
5	EXT	INT
6	BACK PRESS CHECK	INT

•Not available with 2, 7, 8 and 9 spools.

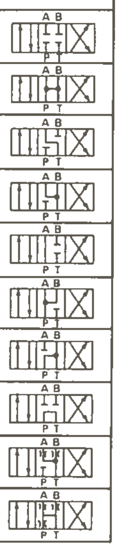
SEAL COMPOUND

OMIT	NITRILE
V	VITON

DESIGN SERIES — 13

SPOOL

CODE	SYMBOL
1	
2	
3	
4	
5	
6	
7	
8 9	
11	
12	



STYLE

	STYLE
B•	SINGLE SOLENOID TWO POSITION, SPRING OFFSET P►A and B►T IN OFFSET POSITION.
C	DOUBLE SOLENOID THREE POSITION, SPRING CENTERED
D•	DOUBLE SOLENOID, TWO POSITION, DETENT
E	SINGLE SOLENOID, TWO POSITION SPRING OFFSET TO CENTER SPOOL SPACER ON SOLENOID B SIDE. P►B and A►T WHEN ENERGIZED.
F	SINGLE SOLENOID, TWO POSITION SPRING OFFSET, ENERGIZE TO CENTER POSITION. SPOOL SPACER ON A SIDE. P►A and B►T IN SPRING OFFSET POSITION.
H•	SINGLE SOLENOID, TWO POSITION. SPRING OFFSET. P►B and A►T IN OFFSET POSITION.
K	SINGLE SOLENOID, TWO POSITION SPRING OFFSET TO CENTER. SPOOL SPACER ON SOLENOID A SIDE. P►A and B►T WHEN ENERGIZED.
P	DOUBLE SOLENOID THREE POSITION PRESSURE CENTERED.

•Available with 1, 4, 11 spools only.

NOTE: 8 spool has a closed crossover.

9 spool has an open crossover.

ATTACHMENTS OR VARIATIONS

OMIT	STANDARD VALVE– NO VARIATIONS
1	CONDUIT ADAPTER
5	SIGNAL LIGHTS
6	MANAPLUG– PLUG-IN
7	PILOT CHOKES
8	STROKE "B" PORT END
9	STROKE "A" PORT END

SOLENOID

Y	115V/60HZ 110V/50HZ
T	230V/60HZ 220V/50HZ
E	24V/60HZ 24V/50HZ
L	6 VDC
K	12VDC
J	24 VDC
D	120VDC

UNIT WEIGHT:
Single Solenoid 27 lbs. (12.2 kg.)
Double Solenoid 28 lbs. (12.7 kg.)

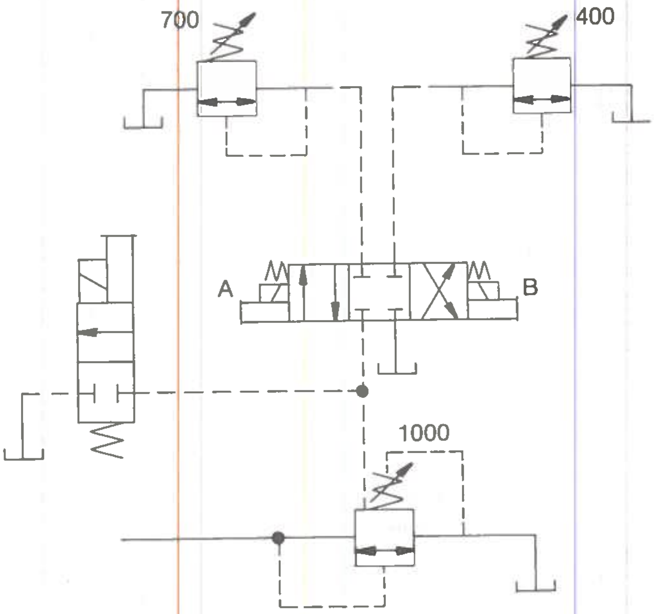
2. If you decided to manufacture your own manifold, what must the microfinish be used and how flat is the surface??

3. For a pressured centered valve, what is the minimum pressure needed to shift the valve??

ASSIGNMENT #17

1. A MRFN12POAFO9AC is set for 1000 psi with 20 GPM passing through it. What is it's cracking pressure?? Use (Figure 7-13) in text. What is its' vent pressure?? Use (Figure 7-18) in text (standard).

2. A MRFN20P6UH08ED is used with multiple pilots and a vent. Fill in the following chart: Use (Figure 7-13) and (Figure 7-18) in text (high vent)

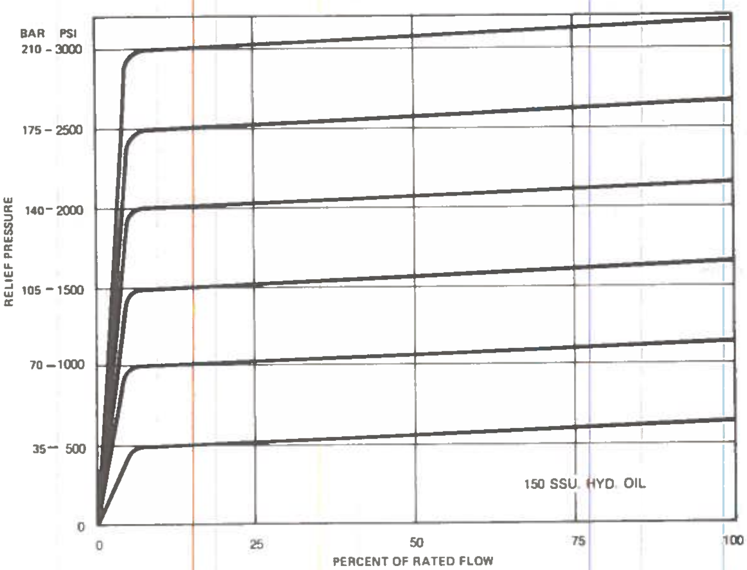


Energized	Pressure	Flow
Solonoid A		60
Solonoid B		40
None		75
Vent		90

Note***Relief valves set at full flow at 90 GPM.

Nominal flow rating is 90 GPM.

3. A RP800S2 must be set so that the system will operate with a load pressure of 1300 PSI. What do you set the relief valve for? Flow is 12 GPM. (Nominal flow of valve is 15 GPM)

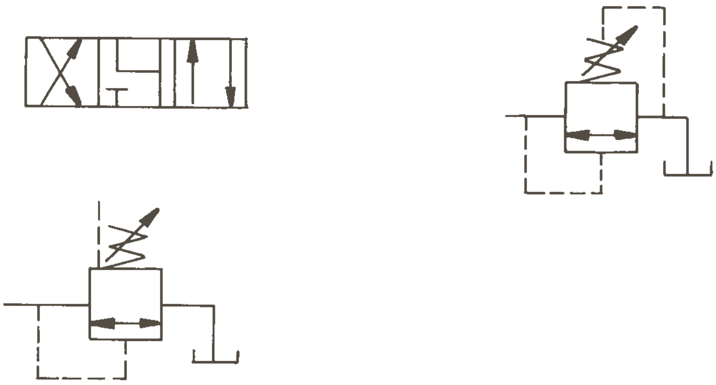


ANALYZING HYDRAULIC SYSTEMS

FINAL EXAMINATION

1. What is meant by venting a relief?? Explain the difference between high and low vents??

2. Connect the components below (pilot operated relief valve; directional control valve, and remote pilot valve) so that 1000 psi is obtained in one position, 700 psi is another, and the pump is unloaded in the final position.



3. When discussing the suction side of a pump, what type of fluid is MORE difficult to suck into the housing; petroleum base or fire resistant?? WHY??

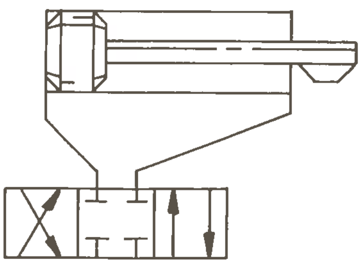
4. Explain the difference between angle and inline checks and where angle checks should be used.

5. What are the four most used center conditions on a three position valve. Explain their advantages and disadvantages.

6. Explain why a closed center valve may cause a cylinder to drift?? If a relief is set for 1500 psi, will the following cylinder drift??

$A_p = 20\ in^2$

$A_e = 10\ in^2$



6000#

7. What are the two methods for charging an accumulator and explain their difference.

8. What is the Beta rating of a filter??

9. What are the three basic types of flow control valves?? How accurate are they to changing surrounding conditions??

10. What are three basic ways a flow control can be used in a circuit?? Explain their accuracies, limitations, and relative efficiencies.

11. How can you control the speed of a hydraulic motor accurately?? Explain why the way you selected was best.

12. When a pressure compensated variable volume pump is fully compensated, is there any flow out of the pump (any outlet)?? If there is, WHY??

13. In a fixed displacement pump, as pressure increases what happens to the flow?? WHY??

14. If you were selecting a cylinder, what would you check for to determine the correct one?? What accessories may you select for a cylinder??

15. What is precharge?? Why is this value selected??

16. What is the efficiency of a typical industrial accumulator system??

17. Pilot operated checks have an amplifier ratio on their pilot sections. What are the common ones??

18. What is meant by decompression poppet?? How can you be sure one will work in the system??

19. What is Cv??

20. What springs are found in check valves??

21. What would double pilot operated check be used for??

22. What is an unloading valve and how is it used in a circuit??

23. What is a sequence valve and how is it used in a circuit??

24. What is a pressure reducing valve and how is it used in a circuit??

25. What is cavitation?? How can it be prevented??

26. What is a counter balance valve used for?? Differentiate between a direct and remotely operated counterbalance valve.

27. How does pressure affect the life of a pump??

30. What is maximum allowable vacuum?? Tell how to use this to determine valuable information.

31. What is regeneration?? Why is it used??

32. Which pressure control valves have external drains??

33. What is different about a tandem center valve compared to any other valve when speaking about shifting characteristics??

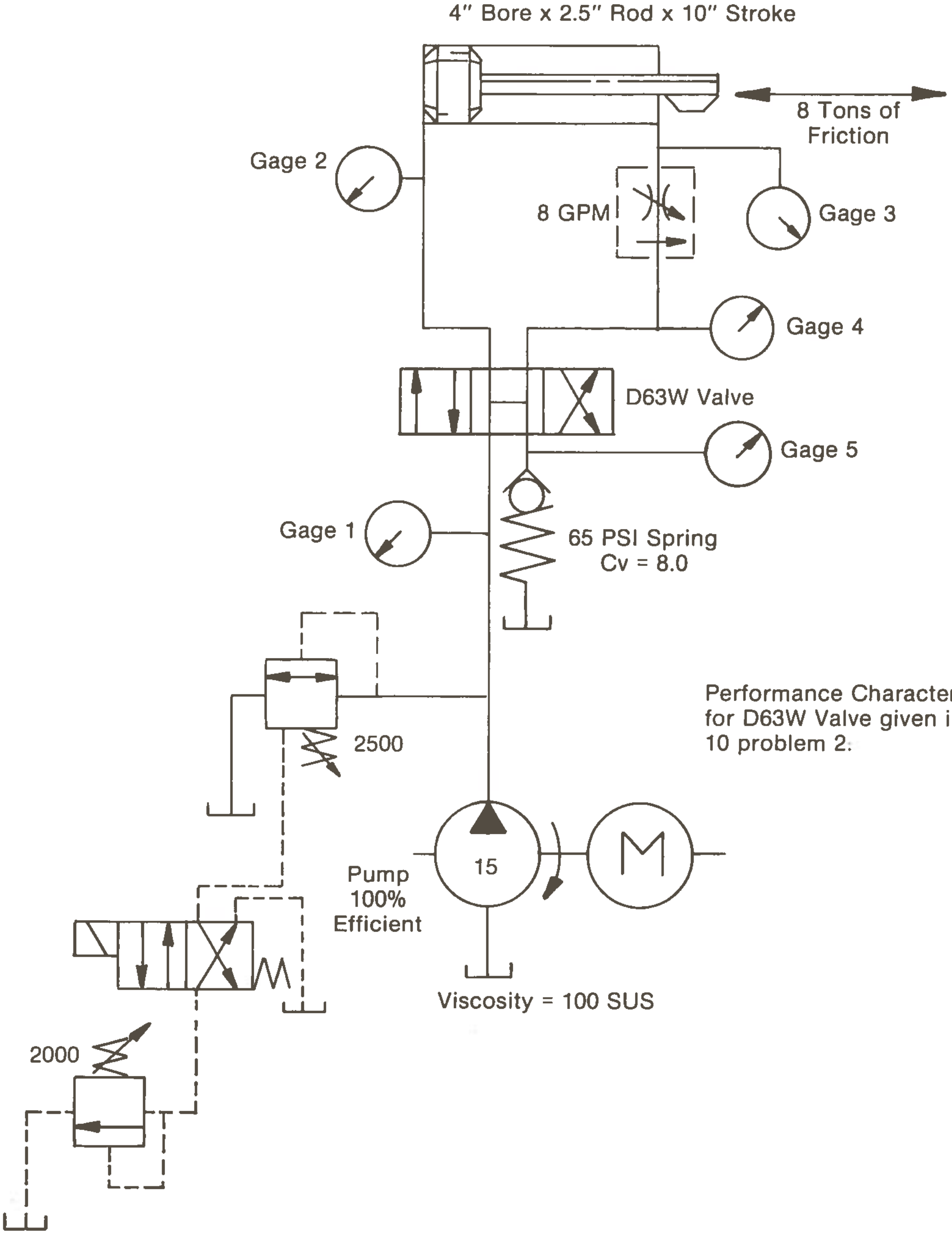
34. If reservoirs are sized too large, what happens to the heat dissipation characteristics?? WHY??

35. What is specific gravity??

36. Fill in the following chart: (Consider the 8 GPM flow control to be able to meter accurately in both directions)

Gage	1	2	3	4	5
Extension	2000				
Retraction	2500				
Idle					

What is the average heat generated in this cycle
if idle time is 10 seconds??



Performance Characteristics
for D63W Valve given in assignment
10 problem 2.

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